EMPIRICAL ANALYSIS OF TWO DRUM WINDING

By

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I would like to take this opportunity to thank those who have made this work possible. First, I would like to thank Dr. Keith Good for being my academic advisor, and giving me the research assistantship that has made this thesis possible. Second, I would like to thank Ron Markum, and Robert Taylor for their assistance in setting up the two-drum winding machine required for this project. Also, I would like to thank the other members of the Web Handling Research Lab for their contributions as well. Furthermore, I would like to thank my parents who provided me with that necessary encouragement that helped me throughout my college career.

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CHAPTER 1

INTRODUCTION

Over the years, the manufacturing industry has seen a need for increasingly faster production of the highest quality products with the least amount of material waste. This need is often the result of a financial interest, consumer need, or an environmental concern. The slightest increase in production often means the potential for more profit through more sales, while the production of an inferior product may often lead to lost sales. The loss of material in the manufacture of products not only means a financial loss, but the disposal of said material often leads to environmental concerns as well. Therefore, many processes in the manufacture of a product are continually improved upon to increase productivity, produce a higher quality product, or to prevent material waste.

At some point in many continuous manufacturing processes is the winding or unwinding of webs from wound rolls. The use of wound rolls is often the easiest way to transport materials, the fastest means of providing material to a manufacturing process, or sometimes the product itself. A few examples of the types of materials used in wound rolls includes: paper, steel, and various types of plastics. The manner in which the product is wound or unwound can affect the quality of the material, or result in the loss of the material itself. For example, rolls that are out of round, or rolls having flat spots, often lead to vibrations in the web film that is wound off the roll. These vibrations can be of sufficient amplitude to cause this web film to break at some point in a given manufacturing process. Sometimes thousands of feet of web may be unwound and undergoing processing, and the breaking of this web can lead to lengthy production delays and material loss.

A good quality roll is one that simply has no defects associated with it. These defects can be any number of things including out of roundness, telescoping, and starring. The quality of a roll is often the result in the manner in which the roll is wound. It has been found that the internal pressure distribution, or radial pressures, is the leading indicator of performance of the roll. These radial pressures can be controlled by the winding tension of the web, or wound-on-tension (WOT).

Controlling the wound-on-tension (WOT) can be done by a number of winding configurations. There are four main winding configurations which include center winding, center winding with a nip roller, surface winding, and two-drum winding (See Figure 1). For center winding the WOT is simply the incoming web tension, but for the other configurations nip mechanics affects the final level of WOT. For center winding with a nip roller and some regimes of surface winding, the relationship between nip mechanics and WOT is known. Currently, for two-drum winding the WOT is unknown.

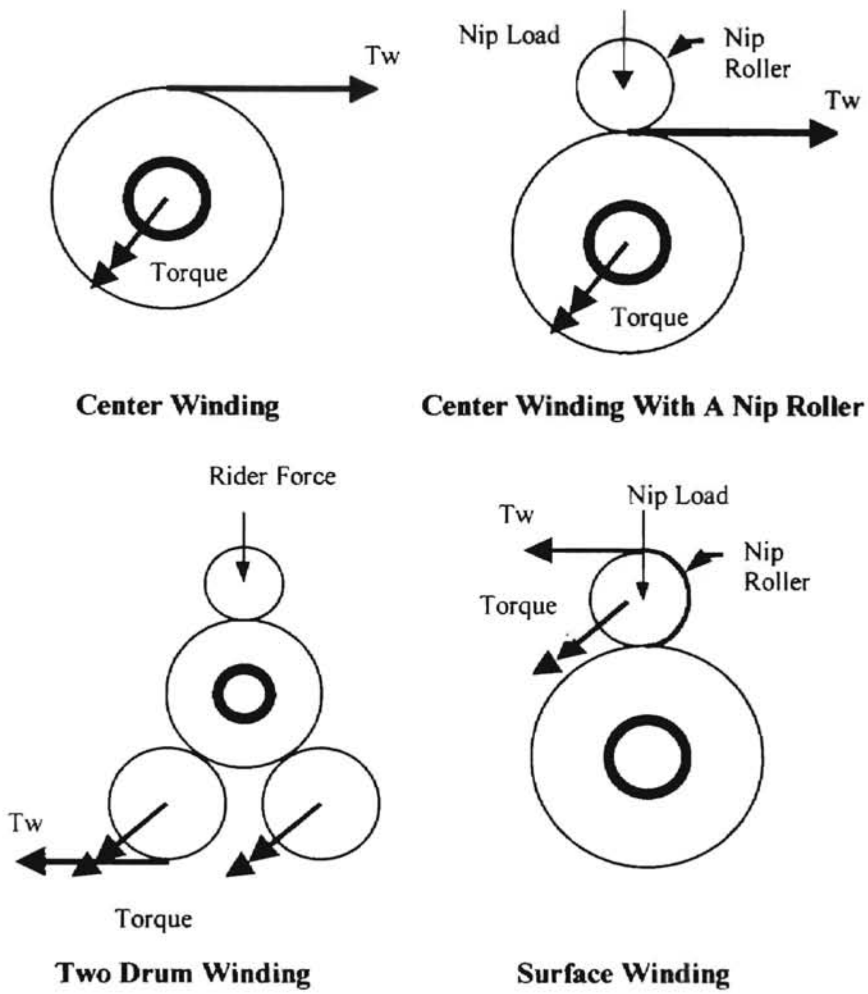

Figure 1: Winding Configurations

The use of the two-drum winder has become a mainstay in the paper industry. The winder is limited by the size of the rolls that can be wound due to the effects of the weight of the roll. When the roll reaches a certain size, the weight of the roll has an affect on the WOT that causes the web to break. It's use of two drums, through which torques are applied, and a rider roll make the mechanics of two-drum winding more complicated than the other winding configurations. The winder turns the roll being wound by applying torques to the two winding drums similar to that of the surface winder. However, unlike surface winding, the added complication of differential torques can be applied to the drums. The rider roll is used to force the winding roll into contact with the two winding drums, and to provide some initial WOT at startup. The load of the rider roll is then decreased as the roll has gained sufficient weight such that slippage of the winding drums does not occur. The winder typically has an unwind stand like those of the other winding configurations that produces an incoming web line tension through a brake.

The effect of nip mechanics on the two-drum winder is something of a black art. For example, it is known that increasing incoming web line tension, torque differential, and rider roll load causes in increase in wound-on-tension, but to what degree is unknown.

The objective of this research is to determine how incoming web line tension, drum torques, and the rider roll load affect the wound-on-tension for two-drum wound rolls.

CHAPTER 2

LITERATURE REVIEW

Many articles have been written regarding two-drum winders, but few articles deal with the mechanics of WOT involved in two-drum winding. It seems that the mechanics involved in two-drum winders are difficult to determine. A mathematical model for center winding with and without a nip roller has been accurately determined. In addition, surface winding model is currently being worked on as of the writing of this report. However, the mechanics of multiple nips with applied torques, has seen in the two drum winder, have scarcely been addressed.

The single most important paper for this project was Hakiel's model [1] for center wound rolls. This model was important as it removes assumptions made by previous authors concerning anisotropic material properties, and allows the radial modulus to be dependent upon radial pressure. It does incorporate the assumption that the wound roll can be modeled as an axisymmetric structure, which implies that little or no slippage occurs within the roll. This model consisted of modeling each layer of paper as a single hoop in a wound roll. For each layer of paper wound onto the roll, another hoop was added to the model. Each of these hoops was then modeled as a second order homogeneous differential equation given by:

$$r^2[d^2(\delta P)/dr^2] + 3r[d(\delta P)/dr] - (g^2 - 1)\delta P = 0$$

[1]

In order to solve this model, two boundary conditions needed to be incorporated, one at the core of the paper, and the other the WOT of the last layer of paper added to the roll.

$$[d(\delta P)/dr]|_{(r=1)} = [(E_t / E_c) - 1 + \upsilon]\delta P|_{(r=1)} \quad \text{At the Core}$$

[2]

$$\delta P = [WOT / s]h \quad \text{Last Layer}$$

[3]

where δP was interlayer pressure at a radius r if s is the outside radius of the roll. Et, Er, and ν were the material properties of the paper of web thickness h. Ec was the radial stiffness property of the core. For center wound rolls, the WOT is simply incoming web line tension, Tw. Hakiel's model [1] has become the basis for the other winding models of center winding with a nip, and surface winding. These other winding models simply find the above WOT as a function of the nip mechanics involved. Thus, if a suitable WOT model can be developed for the two-drum winder, and slippage is assumed negligible, then Hakiel's model [1] could be used.

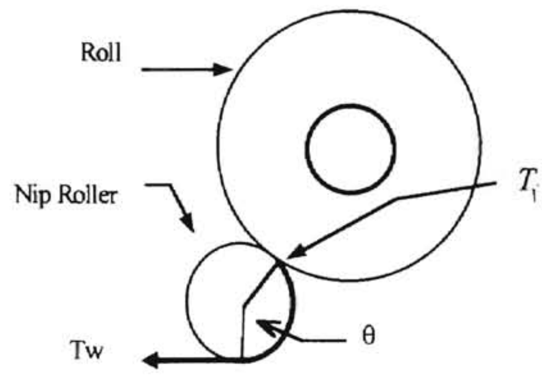

Figure 2: Surface Winding Showing Web Capstan

Pfeiffer[2] was the first to investigate the effect of nip mechanics. He concluded that a wound roll in center winding could be modeled with an infinite radius. This meant that center winding with a nip roller could be modeled as several sheets of paper laid out flat on a table with a nip roller rolled over the top sheet. A force gauge could then be attached to the sheets, and a load could then be measured as the nip rolled down the sheets. From these experiments he concluded that the wound-on-tension, WOT, due to the effect of the nip was additive to the initial web tension. Also concluded from the experiments, was that smaller diameter drums produced higher WOT than larger diameter drums.

Pfeiffer[2] also presented a partial WOT equation describing the incoming nip tension for surface winding. This equation was due to an effect of a wrapped capstan of the winding drum (See Figure 2). This wrapped capstan has the effect of reducing the incoming web tension by the equation:

$$\frac{T'_1}{T_w} = \pm e^{\mu_{k,p-p}\theta} \quad [4]$$

where Tw was the incoming web tension, T'_1 was the incoming nip tension, $\mu_{k,\,p\text{-}p}$ was the paper to paper kinetic friction, θ was the wrap angle of the web around the drum. This equation was derived from solid mechanics theories with a fixed roller. Note, that wrapped capstans exist in two-drum winding.

Good, Wu, and Fikes[3], developed a formula for the WOT for center wound rolls with loaded nip rollers. As stated above by Pfeiffer[2], the effect of the nip on the WOT was additive to the incoming web tension, T_W. This formula is:

$$WOT = T_w + \mu_{k,p-p}\frac{N}{h} \quad [5]$$

where $\mu_{k,p\text{-}p}$ was the friction between the successive layers of paper, N was the applied load to the nip in pli, and h was the web caliper in inches. This formula was obtained by using computational mechanics and verified using an experiment similar to Pfeiffer[2]. Here strips of paper were laid flat out on a table and a nip roller was rolled over them to determine a strain in the paper. This developed formula was then applied to Hakiel's model [1] for center wound rolls, which was then compared to the actual pressures measured in surface wound rolls. With these comparisons, the above formula was deemed correct. Note the above formula does not include a nip drum radius as proposed by Pfeiffer[2]. Also, that the two-drum winder has one undriven roller, the rider roll.

Cai[4] developed a modified formula of Good, Wu, and Fikes[3] for the WOT for surface winding. This formula, based on Pfeiffer's[2] noted effect of wrap capstan as stated above, is:

$$WOT = \frac{T_w}{e^{\mu_{k,p-p}\theta}} + \mu_{k,p-p}\frac{N}{h}$$

[6]

Cai developed this formula in regard to his work with compliant nip rollers. Therefore, he did not do a full study of equation 6. As pointed out by Clark[5] who further studied equation 6, this equation only works for low nip loads and fails to apply for the high nip loads. For high nip loads, a bubble can form in the incoming web as it enters the nip. This bubble represents a web tension of zero for this case, which the above formula does not take into consideration.

A recent work at the Web Handling Research Lab by Steves[6] concerns a surface winding model. He's model relates the velocity differences that occur between the nip roller and the winding roll by:

$$WOT = \frac{V_R - V_N}{V} E_t + Tw$$

[7]

V_R = Velocity of the winding roll

V_N = Velocity of the nip roller

V = Average velocity of nip roller and the winding roll

Et = Tangential modulus of elasticity of the paper

If the

$$WOT \le \mu_{k,p-p} \frac{N}{h}$$

[8]

Then, the WOT is predicted by equation 7. If equation 8 does not hold, then the WOT is simply:

$$WOT = \mu_{k,p/p} * \frac{N}{h}$$

[9]

Because torques are applied through the nip roller, this winding configuration resembles that of the two-drum winder. For each drum on the winder, through which torques are applied, the effect of equation [7] or [9] are expected to be seen.

Rand and Eriksson[7] were the first to show the effects of multiple nip rollers on wound-on-tension. Their experiments were conducted by gluing a strain gauge to the web, and graphing the web line tension as the web passed the rollers during the winding. It should be noted that the backing on which the gage is on has a higher elastic modulus than paper. Therefore, applying strain gages to paper will generate a certain amount of unknown error in measurements.

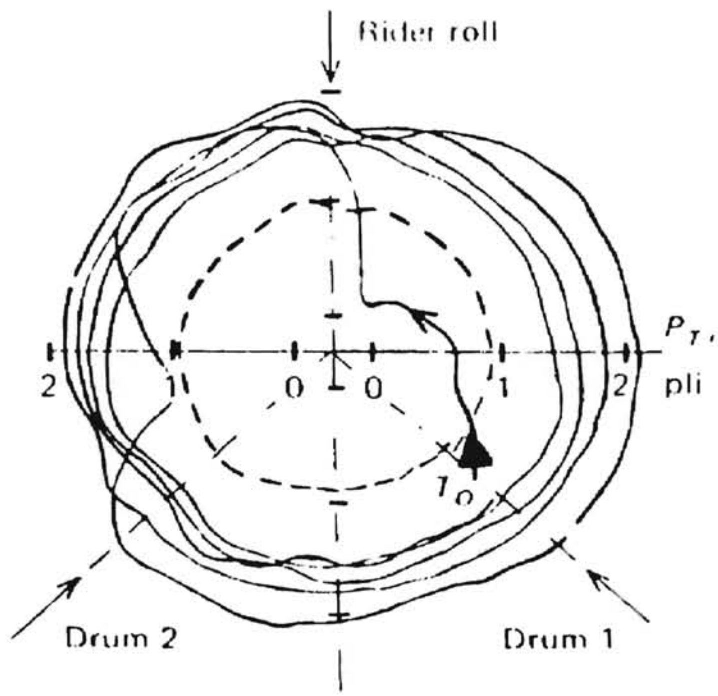

Figure 3: Rand WOT Graph of Two-Drum Winder with a Roll dia. 4 in.

Rand and Eriksson's experiments for a undriven nip roller showed the web tension dropping before the nip and increasing upon leaving the nip. The same tests were also performed for two-drum winders, where a polar graph indicated the changing web tension as the gage passed the different nip loads (See Figure 3 and Figure 4). The tension would slowly decrease after passing drum one and then sharply increase after passing the rider roll, and the effect could be seen again when coming to the second winding drum.

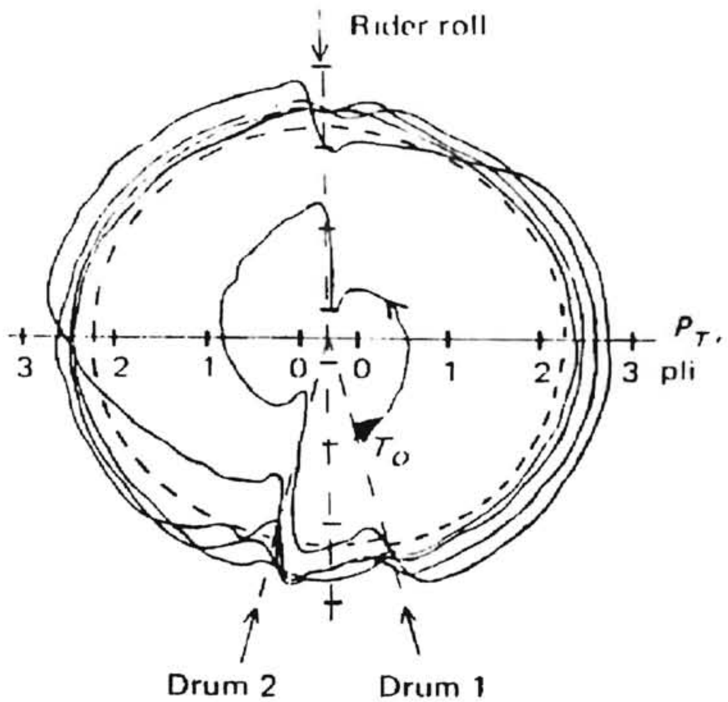

Figure 4: Rand WOT Graph of Two-Drum Winder with a Roll dia. 31 in.

As the roll continually added more layers of paper, the tension would decrease. It could be determined from the plots that the rider roll and drum two contributed most of the WOT for the roll. In addition, as the winding roll becomes larger the angle of the normal reaction forces to the rolls changes, and the WOT becomes larger. Rand and Eriksson[7] did not give any data concerning incoming web tension, rider roll, and torque load used to produce these figures, thus making any firm conclusions concerning nip mechanics impossible. In particular, T_0 is given as the starting WOT, and with the incoming web tension not specified causes difficulty in determining the effects of drum one on the WOT.

Another contribution to two-drum winding was that by Frye[8]. His work was the logical continuation of Pfeiffer's tests [2] of rolling a nip roller over paper laid flat out on a table. Instead of using one roller as in surface winding, he used three rollers as seen in two-drum winding. Each one of these rollers represented one of the two winding drums or the rider roll. In the experiments, each representative roller touched the paper in the order as each drum or rider roll would have touched the paper in two-drum winding. Each load applied to the roller was then determined by the load that would have been applied from each drum or rider roll during actual winding. Differing two-drum winding configurations were tried by either changing the size of, or the inclination of the winding drums. Frye's tests [8] showed that smaller diameter drums, or having the drums inclined, created higher wound-on-tensions.

Frye[8] performed experiments to determine how much each the two drums and the rider roll contributed to the WOT. It could be seen from Frye's[8] results that drum one produced most of the WOT in any given winding configuration. In addition, the rider roll and drum two contributed to the WOT as well, thus indicating that multiple nips are each additive to the WOT. Frye[8] did not take any steps in his experiments to account for the torques applied to winding drums. Because the effects of torques were not taken into account, this gives doubts as to how well Frye's[8] experiments will actually apply to the two-drum winding situation. Furthermore, when comparisons are made to Rand and Eriksson's [7], Figure 3 and Figure 4, it is noticed from these figures that drum one appears to contribute little to the WOT.

A very recent work by Olsen[10] has developed an analytic model for two-drum winding. This model was a dynamic analysis of winding which included the initial effects of high velocity wound rolls. Olsen's[10] derived formula was:

$$\text{WO}\,T = T_1 - \rho \upsilon^2$$

[10]

were T_1 was defined as the WOT, and is the tension in the web just after the outer layer has passed onto the wound roll after exiting drum one, and is given by:

$$T_1 = \frac{1}{2}\left(T_w - T_s + \frac{M_2}{hr_2} - \frac{M_1}{hr_1} \right) + \frac{F_n}{h} + \left(\frac{1}{8}\rho + \frac{I_c}{4\pi s^4} \right) v^2 + \left(\frac{2I_1 / r_1^2 - \Omega}{4\pi ss} - \frac{1}{4}\rho s \right) v \frac{dv}{ds}$$

[11]

where the rolling resistance terms are given by:

$$T_s = \frac{\tilde{M}_{br}}{h * r_r} - \frac{\tilde{M}_{r1}}{h * r_1} + \frac{\tilde{M}_{r1}}{h * s} + \frac{\tilde{M}_{r2}}{h * r_2^{\bullet}} + \frac{\tilde{M}_{br}}{h * r_r^{\bullet}}$$

[12]

The inertia coefficient is given by:

$$\Omega = \frac{\tilde{I}_1}{r_1^2} + \frac{\tilde{I}_2}{r_2^2} + \frac{\tilde{I}_r}{r_r^2} + \frac{\tilde{I}_c}{s^2}$$

[13]

and M_1 and M_2 were the torques applied to the respective drums, and M_{r1} and M_{r2} were the respective torque resistance. Also, r_1 and r_2 were the drum radius, and r^*_1 and r^*_2 were the effect radii given by:

$$\frac{1}{r_1^*} = \frac{1}{r_1} + \frac{1}{s}$$

[14]

Also, υ was the velocity of the web, ρ was paper density, and s was the outer radius of the roll. Where I_1 and I_2 were the moment of inertia for the first and second drum, I_c was the moment of inertia for the core, and F_n was the tangential contact force given by:

$$0 \leq F_n \leq \mu N_1$$

[15]

N_1 was the normal contact force between the wound roll and drum one.

If velocity and resistance terms are assumed to be negligible equation 10 becomes:

$$WOT = \frac{1}{2}(Tw + \frac{M_2}{r_2} - \frac{M_1}{r_1}) + F_n$$

[16]

This equation [16] seems to follow current beliefs about two-drum winding. The formula contains a positive incoming web tension, and a differential torque term which would agree with industry observations. The torques are also divided by the radius of the drums as would be expected from statics. The formula also contains a term for surface windin effects, F_n, in which the effects of the weight of the roll would have an affect here.

Olsen's[10] work is purely analytic, with no comparisons made with experimental data. Furthermore, all of Olsen's[10] work is based upon finding T_1, the tension of the web after passing the first drum, as the WOT for two-drum winding. As mentioned before with Rand and Eriksson's[7] experimental data, the WOT after exiting drum one is not significant. Furthermore, the effects of decreasing web tension before entering a nip, as seen in Figure 3 and Figure 4, are not taken into account in Olsen's work. Therefore, although Olsen's analytic formula follows established beliefs about WOT affects for two-drum winding, Olsen's formula gives contradictions when compared to the physical data of Rand and Eriksson[7].

Most literature related to two-drum winding deals with the setup and operation of the machine. One such work on the background of two-drum winding is that by Frye[9]. A typical setup for a two-drum winding machine is based on one or two motors to control the differential torques to the drums. These motors are connected either through some type of gearbox, belts, or directly attached to the winding drums. These motor/motors can be permanently set, or adjusted, to produce a difference in speeds or torques between the drums. Typically, the two-drum winder runs at high speeds so the first drum is grooved to prevent a build up of air, and the second drum is coated for traction. There are two methods for bring the incoming web into the winder the between-drums, and front-drum method (See Figure 5).

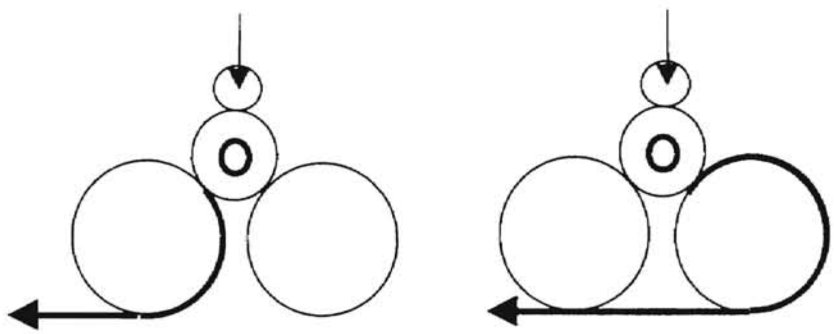

Figure 5: Web Path For Two-Drum Winder

The between-drum method wraps the paper around the first drum, following a path that leads the paper between the drums, as done for this project. The front-drum method wraps the paper around the outside of the second drum of the machine. The rider roll is used to provide some initial WOT at the startup of the machine, and to put the winding roll into contact with the winding drums. This rider roll is typically preprogrammed to be brought up during winding by following some predetermined decreasing load curve. The load to the rider roll is typically controlled by actuating cylinders.

An issue with surface winding, and center winding with a nip, was whether the incoming web tension was additive to the WOT created by the nip roller. As demonstrated by Steves[6] model for surface winding, the incoming web tension is additive to the WOT until the nip roller begins to slip. After the nip begins slip, the WOT is then a function of the kinetic friction and the nip load. An important issue for two-drum winding was whether the WOT is the additive effects of Tw, the two winding drums, and the rider roll, or simply the result of the nip with the highest load. As seen in Figure 3 and Figure 4, it is difficult to determine the additive effects of the nip rollers due to drops in WOT before entering each of the nip rollers. If the WOT in two-drum winding was the result of the nip with the highest load then, the first drum would produce the most WOT. Then subsequent nip rollers with lower or equal loads would mostly likely slip before contributing to the WOT of the roll. This being the case, WOT contributions from the rider roll and drum two should not be seen. Inspection of Figure 3 and Figure 4 shows the opposite with drum one contributing little to the WOT and the rider roll and drum two creating most of the WOT. Frye[8] would suggest that each nip in two-drum winding was additive with the first drum producing most of the WOT and the succeeding nips contributing a smaller proportion to the WOT. The fact the Frye[8] suggests drum one produces most of WOT contradicts the graphs produced by Rand and Eriksson[7]. Olsen[10] is the only work that makes any attempt at an analytical solution for two-drum winding. But, Olsen's[10] work is based upon the WOT after the first drum, and equates this web tension to the WOT. Again, Rand and Eriksson data would suggest that the WOT after drum one is not significant.

The objective of this research is to determine how each of the four winding variables, incoming web line tension, the two drum torques, and the rider roll load affect the wound-on-tension for two-drum wound rolls. To achieve this objective, a two-drum winding machine is setup for experimentation. Then, a number of different experiments are performed by changing one these four winding variables and determining the resulting changes in radial pressure. These radial pressures, measured from the rolls wound in the experiments, were then related to an estimated WOT determined by a graph produced from Hakiel's center wound model [1]. As a conclusion to this project, an empirical model based on surface winding effects, and on the four winding variables predict the estimated WOT. This empirical model will show which of the four winding variables has an effect on WOT. Furthermore, a second empirical model is developed based upon the effects seen in Rand and Eriksson's polar plots. Also, Olsen's [10] two-drum winding equation is compared to the estimated WOT.

CHAPTER 3

EXPERIMENTAL SETUP

A two-drum winding machine was set up to represent the conventional two-drum winder (Figure 6). Two steel drums were used as the winding drums, each powered by a 5-hp. electric motor. These motors are, in turn, connected to force transducers that measure the torques via signal conditioning hardware and a data acquisition and control system. It should be noted that the first motor rotates at preset constant speed, controlled by the motors' controller, while the computer system controls the amount of desired torque to the second drum. This is typical in the industry, as the velocity of the web process dictates the speed of the winding machinery. The motor on the first drum provides whatever torque, within bounds, is necessary to maintain constant velocity. The second drum may assist the first drum in maintaining constant velocity. A rider roll comes down from the top, and is counter balanced by weights, and is used to hold the core in between the two drums. A force transducer is used to measure web line tension, which is used as feedback to a brake controller on the unwind stand. Thus, closed loop tension control is achieved within the web prior to winding.

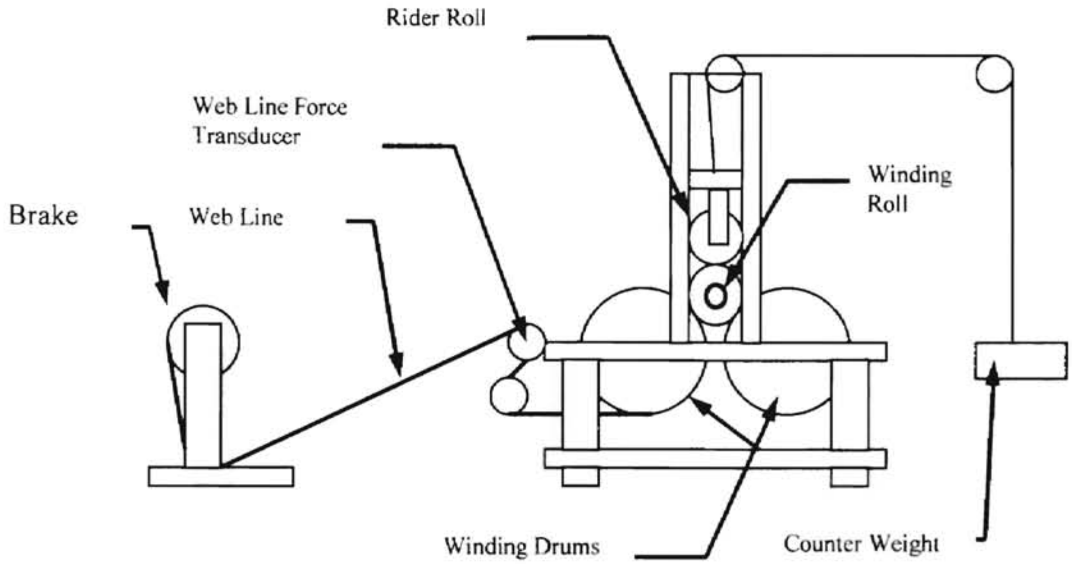

Figure 6: Drawing of Two Drum Winder Setup Side View

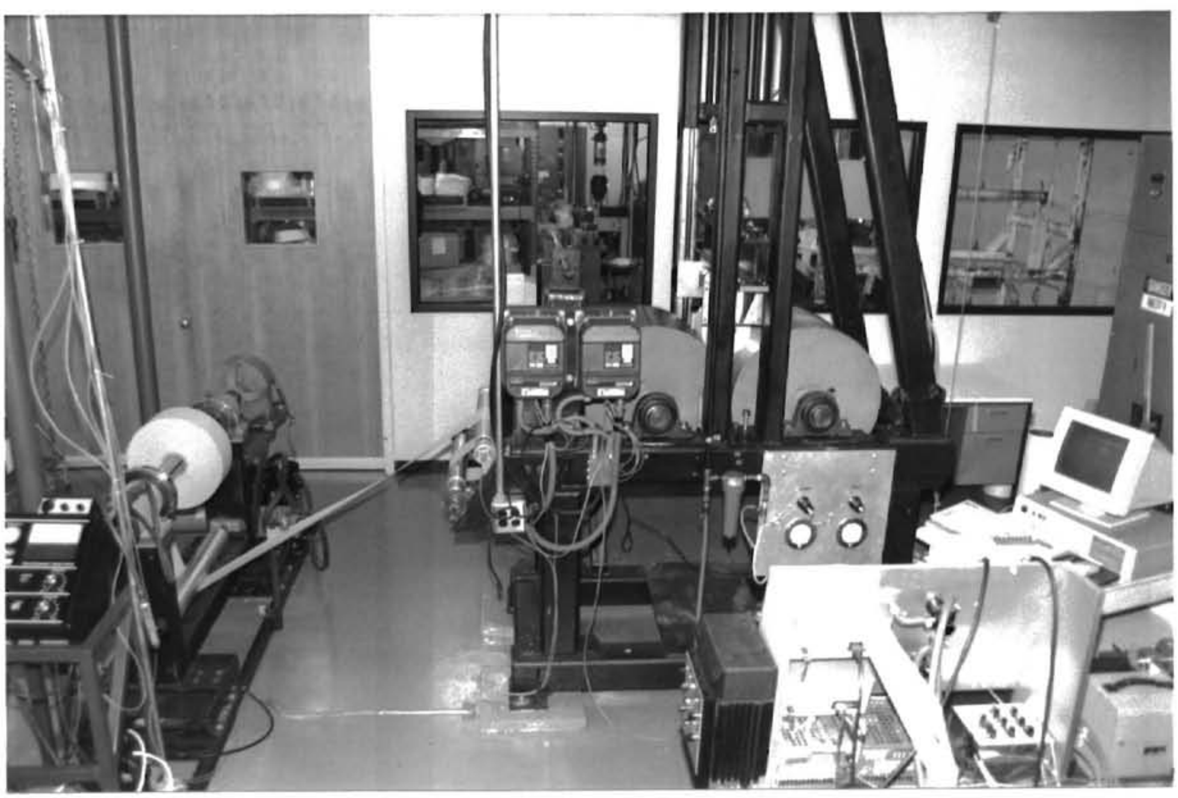

Figure 7: Photograph of Two Drum Winder Setup Side View

MACHINE SETUP

Machine Dimensions

The dimensions and the placement of the drums were important when deriving modeling equations for the two drum winder(See Figure 8 and Figure 9). The winding drums are 24 inches in diameter, and have recently been resurfaced with 64 micro inch finish. The centers of these winding drums are exactly 24.5 inches apart, and the alignment of these drums are within one thousandth of inch between the ends of the shafts on the drums. The rider roll is 6 inches in diameter, and comes down centered, as it deploys between the two winding drums.

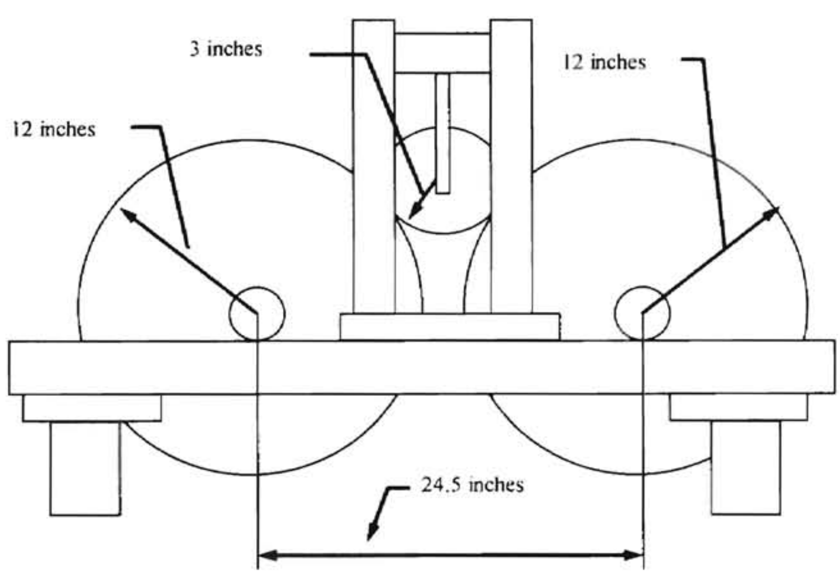

Figure 8: Dimensions of the Two-Drum Winder

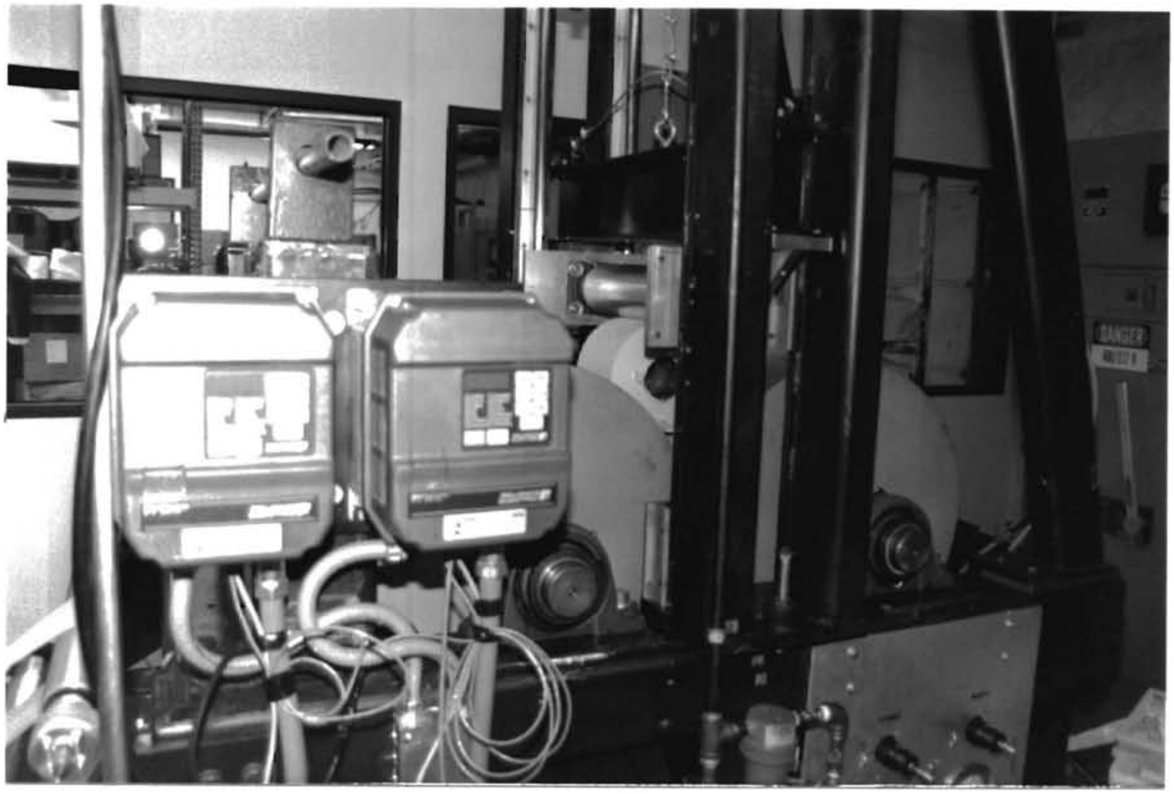

Figure 9: Photograph of Two-Drum Winder Close up of Side View

It should be noted that nothing holds the core for this winding configuration. Therefore, the core is free to move along the length of the drums. Due to this fact, the maximum pile height of paper is 3 inches (from the outside of the core to the final radius of the winding roll). After 3 inches of pile height, the core will begin to move along the length of the drums causing major telescoping of the roll.

Motors/Controllers

Two new motors were installed for this project for purposes of powering the winding drums. The motors used were two Reliance Electric Motors both supplied with Reliance Electric GV3000 A-C Drive controllers. These motors were capable of producing 5 horsepower at 180 inch pounds of torque for a period of one minute, and a maximum of 270 inch pounds of torque before the controllers reset the motors, shutting down the machine. These motors were supported directly off the shafts of the winding drums; the only thing preventing the motors from freely turning was the force transducers. Each of the controllers supplied were capable of controlling the various parameters of each of these motors such as speed, torque, and acceleration. Operation of each controller could be done manually via a keypad, or from an outside signal such as from a data acquisition and control system.

Torque Measurement

Measuring the torque on the winding drums was accomplished using force transducers connected to the motors(See Figure 10, Figure 11, and Figure 12).

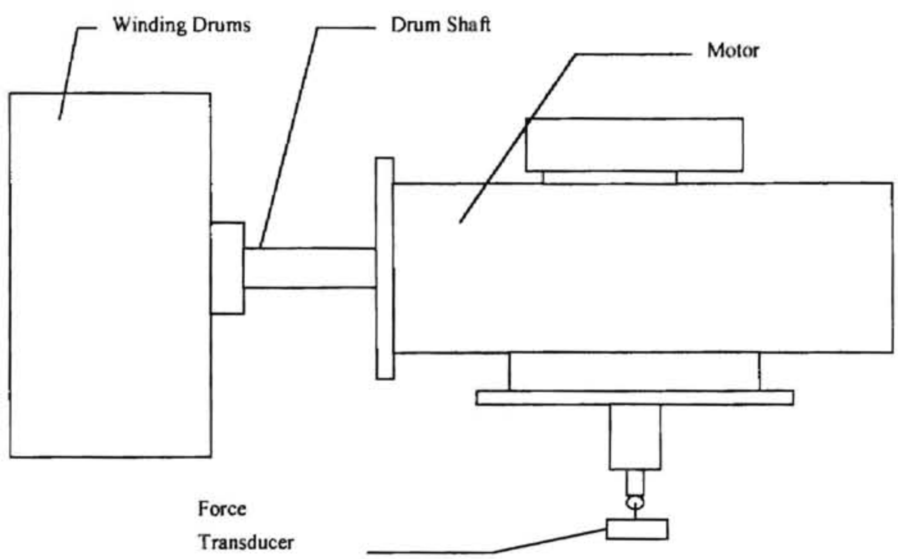

Figure 10: Drawing of Motor and Force Transducer Setup Side View

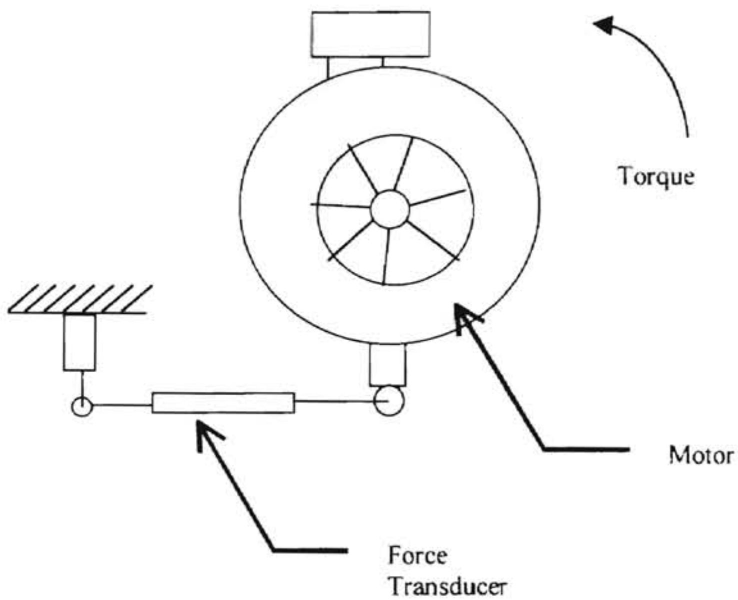

Figure 11: Drawing of Motor and Force Transducer Setup Back View

Figure 12: Photograph of Force Transducer Back View

One side of the force transducer was connected to a small shaft that is bolted to the mounting plate of the motor. The other end of the transducer was connected to a table as a ground. The motors were supported only by the shafts of the winding drums, and only the force transducers keep the motors from rotating. With this arrangement, any torque input by the motor to the drum is measured directly by the transducer. If the load transducer were to fail, the small shaft attached to the motor would act like a safety pin that will restrain the motor from rotating by hitting the table.

Another method for measuring torque would be to measure the current being provided to the motor, and developing a torque versus current relationships. Nevertheless, this is not nearly as accurate as the measurement system designed. Both methods ignore the torque associated with the rolling resistance of the bearings that support the drums.

Data Acquisition System

The controller to the first motor was set to maintain a constant speed of 50 ft/min on the first drum for all experiments. This speed can be controlled only by the controller, via it's keypad. The torque from the first motor is then read by the data acquisition system which is then multiplied by the desired difference of torque. This multiplied torque is then sent back to the second controller that is used to control the torque to the second motor.

This data acquisition and control system consists of two National Instruments boards: a Lab-PC+ board, and a SC-2043-SG board. The SC-2043-SG board was used to excite, measure, and amplify the strain gage bridges in the force transducers. The Lab-PC+ board was a data acquisition board used to read the measurements from the SC-2043-SG board. The signals to these boards were set up for non-referenced single ended signals. The signals going to the second controller controlling the torque to the second motor were set up for 0-10 volts, while the signals measuring the torques were set up for ± 5 volts.

The software to run these boards was Labview by National Instruments (See Appendix C). This program operates like an electrical circuit drawing board, and provides the various functions needed to run the boards. The program written for this project measures and controls the torques for this project. At the end of the execution of the program, all torque data is saved to a desired file.

Rider Roll

The rider roll is used to supply a downward, sometimes varying, force on the winding core during initial winding. However, for purposes of simplification, the rider roll load was always held at a constant load throughout the entire winding process in all experiments. This rider roll is guided by a slider guided and moves in vertical direction to the machine when fully down, and seats directly in between the winding drums. A cable pulley system was set up to supply a counter balance weight to the weight of the rider roll.

One of the variables to this experiment was the force supplied to the winding core from the rider roll. After the rider roll has been balanced simply taking on and off counter weights can give the desired rider load.

Brake Control

A braked unwind-stand on this system controls the incoming web line tension for the two-drum winder. The brake is located on the unwind stand for the roll, and has it's own controller. A force transducer measures the incoming web line tension, and is used as feed back for the brake control. A minimum of 1 pli web-line-tension was required for control purpose for the data acquisition system. Web line tensions of less than 1 pli cause the torques to the drum to oscillate by some 100 inch pounds. This effect is thought to be the result of the gain settings set on the motor controllers. If the gain settings were set too high, it would cause an overshoot on the desired torque for the motor. If the first motor overshot the desired torque then the torque to the second motor may compensate by reversing itself. This effect would continue, alternately switching between the drums with the system trying to equalize, but still overshooting the desired torque. The result would appear as oscillations of torques on the computer. This matter was not fully investigated, but is believed as the most likely culprit for the oscillating torques. Also, incoming web tensions greater then 4.4 pli will cause one of the motors to exceed 180 inch pounds of torque. After one minute above 180 inch pounds, the motor will shut off causing an unknown change in the pressure profile of the winding roll.

PULL TABS

Pull-tabs were used in the experiments to measure the radial pressures that develop in rolls. These tabs were inserted during the winding process typically at the brake. The force required to dislodge the tab is related to the pressure applied to the tab. Seven to six pull-tabs were inserted evenly into every wound roll to establish the radial pressure profile for that roll. The making and calibration of the pull tabs is described in appendix A.

LIMITS ON WINDING

There are three limitations of the two-drum winder. The first limit was that only 180 inch pounds of torque can be supplied by the motors. When this maximum torque was reached for a period of one minute, the motor would shut off to prevent motor overheating. Several things or combinations can cause this limit to be reached:

- Incoming web tension (Tw) exceeding 4.4 pli.
- Creating too large of a difference in torque's between the two drums.
- Too much rider roll force.

The second limit to the machine is the size of roll that the machine can wind. After 3 inches of pile height of paper have been added to the winding core, roll telescoping begins to occur. The third limit is that the incoming web line tension has to be maintained above 1 pli. As described previously, web line tension less than 1 pli causes the torque measurements to oscillate.

CHAPTER 4

EXPERIMENTAL PROCEDURE

There are a number of things that must be done before and during the winding of the rolls. Before any rolls are wound, several things have to be calibrated including:

- Force transducer measuring incoming web tension.
- The torque, or signal sent to the second controller.
- Force transducers measuring torques at drums one and two.
- Rider roll load.

After the system has been calibrated it is a simple matter of turning on the data acquisition system, setting the correct web line tension, and setting the correct rider roll load. While the machine is running the pull-tabs are then inserted.

CALIBRATING THE MACHINE

Calibrating Torques

Calibrating the torque force transducers was done in the same manner as that of a standard strain gage. Typically, this calibration procedure was done with the motors, controllers, etc. turned on. The fans on the motors produce a slight amount torque, and the motors and controllers produce a certain amount of noise that is accounted for in the calibration. All calibration adjustments were made by the computer via Labview software.

The first procedure done was that each transducer's measurement readings were zeroed out with no load. Second, a bar with a hole at the end of the bar was bolted on to a mounting bracket on the motor. This bar has a length of 13 7/8 inches from the centerline of the motor to the hole at the end of the bar. This bar when balanced about the centerline of the motor has a weight of 4/5 pounds at the hole at the end of the bar. A lead weight of 13.0 pounds was then placed in the hole at the end of the bar. This bar and weight has a total weight of 13.8 pounds at a distance of 13 7/8 inches from the centerline of the motor, thus producing a torque of 191.5 inch pounds on the transducer. Third, the incoming signal read from the data acquisition board was then adjusted such that the torque measurement reads 191.5 inch pounds. After the adjustment, the load was removed and the torque measurement was checked to see if still zeroed out. If not, the above procedure was performed again.

Due to some noise in the system, on the order of ± 5 inch pounds, an average value of torque was calibrated. This was, typically, done by doing the above procedure and allowing the data acquisition system to take measurements for a period of 15 to 30 seconds. Upon stopping the program, the data acquisition system wrote the measured torques values to a file. This file was then loaded up in a spreadsheet where an average torque value could be calculated. After the average value was found, the torque measurement was then adjusted accordingly.

Calibrating Rider Roll Load

For the correct rider roll force to be applied to the winding roll the rider has to be properly counter balanced. This was done by placing a piece of plywood across the winding drums where the rider roll would normally sit. A weighing scale was then placed on the plywood, and the rider roll was made to come to rest on top of the scale. It should be noted that there was a large amount of static friction associated with the slider guides on the rider. The static friction in guide caused unrepeatable measurements in weight of rider roll to the tune of ± 5 pounds. The rider roll has to be made to vibrate to put the rider in a kinetic friction mode rather then a static mode. This vibration can be done by manually tapping the pulley cable allowing consistent measurements to be made by the weighing scale. The rider roll was then balanced by adding weights making the scale read zero while tapping the cable. It should also be noted that the winding rolls are not perfectly round. The roundness of the rolls caused a slight vibration of the rider roll during winding causing kinetic friction in the slider guide.

Calibrating Web Line Tension

Calibrating web line tension was typically done through the brake control panel. The web line force transducer is up the web line, and is used as feed back for the brake control. This force transducer was calibrated the same as a strain gage or the torque transducers. With no web line tension, tension was zeroed out at the brake control panel. A string was then used to follow the path of the web line, and a known weight of 13.0 pounds was placed at the end of the string. This weight produces 2 1/6 pli of web line tension, and the brake control was then adjusted to read 2 1/6 pli. After calibration, the weight was removed and the web tension was checked again to see if properly zeroed out. If not, the above procedure was repeated as necessary.

OPERATION OF MACHINE

Operation of the two-drum winding machine was simple. Before any winding was done, all force transducers etc. were calibrated. Typically, calibration of the machine was re-performed after every five to seven rolls wound. The experimental requirements determined how the winding variables were to be adjusted. The winding variables included the incoming web tension(Tw), rider roll force, and percent difference in torques between drums. After these variables had been adjusted to their desired settings, the machine was turned on, and the winding begun. For all rolls wound in this project, the winding speed was set at a constant 50 ft/min, and the total web length on each roll was 1,500 feet of paper. During the winding, pull-tabs were inserted.

Placement of gages

In order to get a good profile of the radial pressures for each roll, the pull-tabs were evenly spaced on the web line. A series of seven pull-tabs were made and calibrated as described in appendix A. The first pull tab was placed on the unwind roll at the brake before winding begins, allowing one tab to be close to the core. The rest of the pull-tabs were evenly spaced at 250, 500, 750, 1000, and 1250 feet. The seventh pull tab became the first pull tab, placed at the brake before winding, when the roll was reused. Each of these pull-tabs was taped onto the roll. Because of the tape, removing the pull-tabs from the rolls is difficult, and sometimes damaging to the tabs. Due to this fact, after tabs were inserted into roll, the rolls were often reused two to three times.

Pressure Profile

After the pull-tabs have been wound into the roll and winding has stopped, the tabs were pulled to give the roll's pressure profile. Each tab was pulled three times with a force gage and the results compared to that tabs calibration curve to determine the average radial pressure for that tab. The location of each pull-tab was then measured from the outside of the winding core to the tab by use of a caliper. A plot was then made of all pull-tabs measuring pressures versus their location from the core. This plot gives the radial pressure profile for that roll.

Possible Experiments

As already mentioned, there are four variables that can be changed for any given experiment, incoming web tension(Tw), rider roll force, and the two torques to the drums. For each of these variables, an experiment was set up by incrementing one of the variables while holding the rest of the variables constant. Each one of these experiments consist of five to six wound rolls each showing the effect of changing one of the variables. The pressure profiles from each of these rolls from a given experiment were then shown as graphs given later.

The range over which these variables could be changed was constrained by the machine limits. The web line tension had to remain within the range of 1.0-4.4 pli to keep the measured torques from the two drums form oscillating, and to keep the motors from overloading. The web line tension was set in increments of 1 pli starting at 1 pli and ending at 4.4 pli. No well defined limits were reached for rider roll force, as the incoming web tension has a bearing here. An increasing rider roll force will decrease the maximum web line tension that can be applied. For this project, the rider roll loads successfully tested included: 1, 3 1/3, 6 2/3, and 10 pli. The percent difference in torques can be adjusted to any desired range. It should be noted however that the desired torque difference was never met. Slippage, or some other aspect of the machine, caused the desired torque difference of 100 percent to become something closer to 30 percent. Typically, the percent difference in torques were changed in increments of 10 percent.

CHAPTER 5

RESULTS

The experimental results for the two-drum winder are shown as graphs of radial pressure profiles of rolls wound for a given experiment. These graphs are produced from pull-tab measurements that give these radial pressure profiles from a given roll in any given experiment. A series of these rolls were wound by incrementing one of the four winding variables, and then plotting their radial pressure values to form a graph. Each one of the graphs represents how incrementing one of the winding variables changes the radial pressure profile. Estimations of these radial pressure profiles are then determined form a graph produced by Hakiel's model [1].

WOT PREDICTIONS USING HAKIEL'S MODEL

For purposes of estimating the wound-on-tension(WOT), Hakiel's model [1] for center winding was used. Pull-tab data shown later in these results exhibit flat plateaus in radial pressure. Experience with center winding has shown that these plateaus are exhibited only when the WOT is constant throughout the wind. Existing center-winding models are employed. For each roll an estimated WOT was made using the two-drum estimation WOT graphs (See Figure 13 and Figure 14). A program, or a script file in C has been written based on Hakiel's center-winding model, and is given in the appendix B. This script file when given a WOT for a given roll will return a predicted radial pressure profile. With the material properties assumed constant from roll to roll, a plot of the average radial pressure profile versus WOT was produced using this script file. With these graphs produced, the central pull tab data was averaged to find an average radial pressure profile for a given roll. With the averaged pressure profile determined, the WOT for the roll is extracted from the graphs.

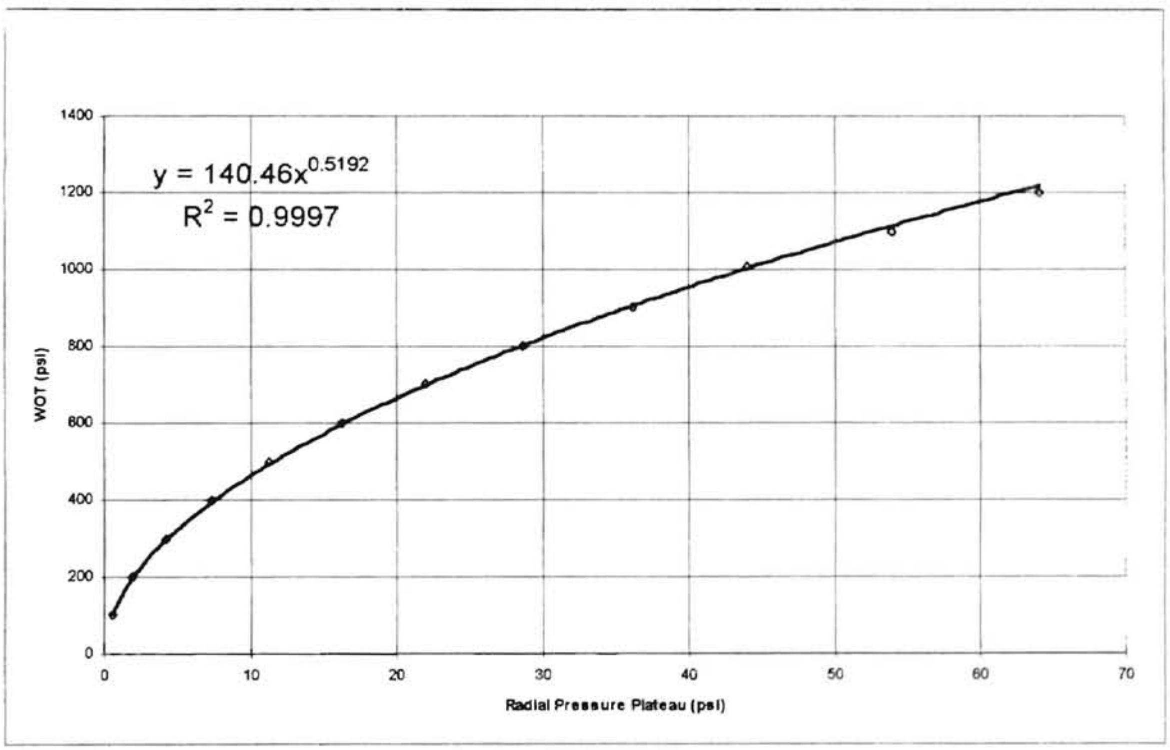

Figure 13: Estimation of Two-Drum Winder WOT in psi.

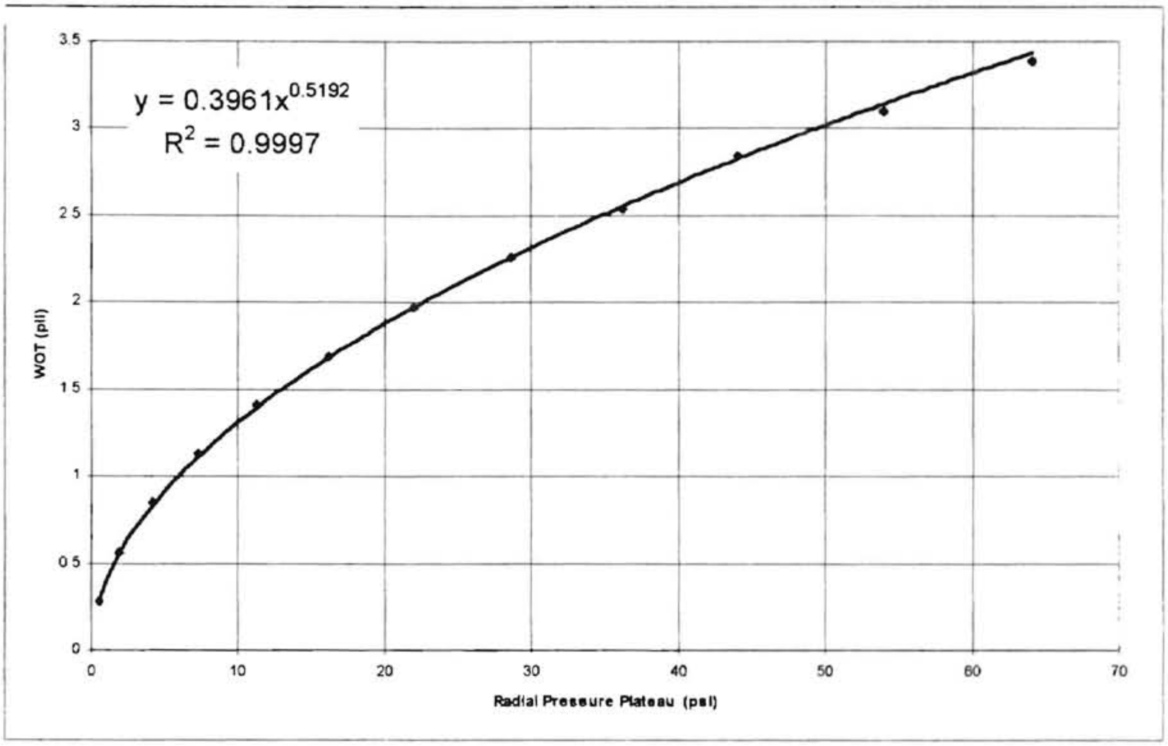

Figure 14: Estimation of Two-Drum Winder WOT in pli

Material Properties

Before Hakiel's model [1] could be used, material properties of paper and the core used had to be determined. These material properties are needed for calculations used in Hakiel's model [1]. Due to previous research done in the lab on other projects, the properties of the paper used, newsprint, and the steel cores used were well known.

The type of paper used for this project was newsprint and its material properties are as follows. For newsprint the modulus of elasticity in the radial direction, Er, was 41*pressure(psi). Er for newsprint is a function of pressure being applied in the roll, and this value for Er is good for pressures up to 100 psi. The modulus of elasticity in the tangential direction for newsprint was 490 kpsi.

There are two friction values associated with newsprint that are of interest, the static paper-to-steel friction, and kinetic paper-to-paper friction. The paper-to-steel static friction, $\mu_{s,\ s\text{-}p}$, for newsprint is 0.25, where the paper-to-paper kinetic friction, $\mu_{k,\ p\text{-}p}$, is 0.19. Neither of these values are required for WOT estimates, but both values are required when deriving a WOT equation for two-drum winding. Therefore, these values are not used in this section, but are used in the "Discussion" section of this report.

The weight of the paper on a given finished wound roll is 6 ½ pounds. This weight was then divided by the width of the roll, putting the weight into terms of pli as required. This weight per length of roll was again divided by the total feet on the roll, 1500 feet. This gives the weight increase in terms of pli for every linear foot of paper added to the roll.

Roisum[11] found the core stiffness, E_c, for the steel cores used for this project to be 3,000 kpsi. However, it should be noted that this stiffness can be adjusted to a much lower value to account for slow increases in WOT upon start up of the two-drum winding machine. A typical center winding prediction of a radial pressure profile shows a high pressure at the core that then drops off to a plateau. Due to the slow increase of the desired WOT by the two-drum winding machine, the radial pressure measurements at the core for two-drum winding are much lower then those predicted by Hakiel's model[1]. Therefore, to get better matches in radial pressure profiles, Ec is adjusted downward to a range of 80 to 40 kpsi for any radial pressure estimates for two-drum winding. A point needs to be made here, a value for Ec is required for Hakiel's model [1], but this value has no significance in WOT predictions. E_c is lowered for purposes of a closer match in estimates in the graphs. The WOT of interest, was the WOT after the machine has reached steady state values in winding conditions. Therefore, the value for Ec has no real meaning, or bearing in this report, but is required by Hakiel's[1] model. The outside radius of the steel cores is 1.7 inches, the inside radius was 1.5 inches, and the core has a weight of 3.5 pounds.

MACHINE OPERATION NOTES

Two notes regarding machine operation have to be made at this point, regarding differential torque control. The first winding drum is speed controlled, and therefore its controller supplies whatever torque is necessary, within its capacity, to maintain a set-point speed. The method for controlling torque to the second drum was based upon the multiplication of the measured torque of the first drum. Therefore, the torque to the first drum is considered the primary torque or the base torque for this project. Hence, the torque difference formula used for this project is:

$$Torque\_Difference = \frac{T_2}{T_1} * 100$$

[17]

where T_2 is assumed to operate at a percent difference from T_1. A true torque difference formula should be:

$$Torque\_Difference = \frac{T_1 - T_2}{T_1 + T_2} * 100$$

[18]

where the total torque is used as the base. This formula [18], is not used for this project, equation[17] is used.

Another note should be made regarding the desired torque difference to the one actually received. The user controls the torque via Labview software that uses equation [17]. By using the torque measured from the force transducer, equation [17] gives a drastically different value for the torque difference. For example, the set torque difference was set at 110 percent while the measured difference may be closer to 40 percent. Throughout the course of the experiments, it could be seen that the torques on the drums, slippage of the drums, rider roll load, and incoming web tension are dependent upon each other. The exact relationship of these variables is unknown, and is thought to have a significant bearing on the above torque difference discrepancy. Therefore, knowing that the measured torque differences are more accurate, only the measured torque differences are presented in the graphs.

A check was made to determine if the torques stay at the same values throughout the winding process. Changing values of torques would have an affect on WOT predictions. For various rolls, the torque data file saved by Labview was checked from beginning to end. No noticeable changes in torque could be found. Therefore, for purposes of this project the torques are considered constant throughout the wind.

The rolling resistance of the bearing and the rolls were also checked for this project. The average torque required to turn each of the drums with no outside load, without winding a roll, was approximately 2 inch pounds. This is the torque required to overcome the rolling resistance in the bearings, etc. The torque measurements vary by some $\pm$ 5 inch pounds for this machine. Therefore, this rolling resistance is approximate, and is presented to give some idea of the magnitude of the resistance. The rolling resistance of a paper roll was checked too. This check was done by inserting a pre-wound roll into the machine and allowed to free roll (no incoming web line on the machine). Torque was only applied to the first drum with differing rider roll loads (See Table 1).

Rider Load (pli)	1	3 1/3	6 2/3	10
Torque 1 (in lbs)	15	17.8	26.7	45.3

Table 1: Rolling Resistance of Rolls at Differing Rider Loads

RESULTS

The results shown are in the form of radial pressure profiles as measured by pull-tabs. Each one of these graphs is an experiment showing the changes in radial pressure as one of the winding variables is increased.

Figure 15 is a check on the repeatability of the base line. This graph was done to check for the range of error that may occur for the preceding results. All winding constants for this experiment are held constant from roll to roll. The difference in pressures between the four test runs was approximately 5 psi. Thus it appears that results may be accurate to within $\pm$ 2.5 psi. The torque data for roll 4 was lost due to a computer malfunction.

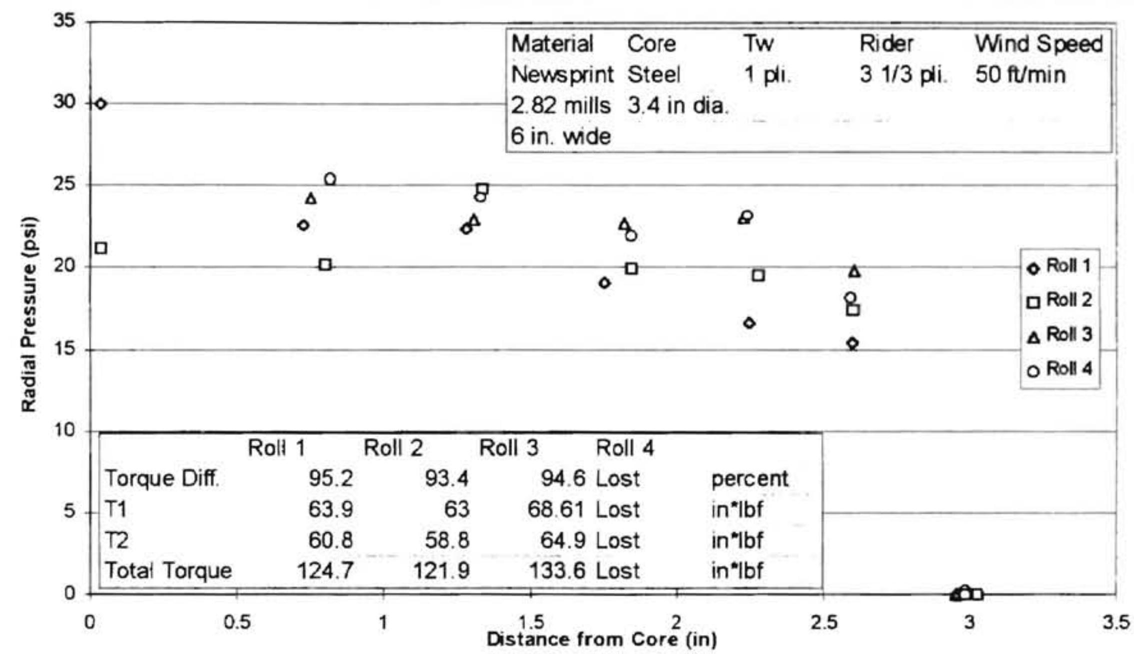

Figure 15: Base Line Repeatability

Figure 16 and Figure 17 show how changing differential torques effect the radial pressure within the wound rolls, at rider roll loads of 1 and 3 1/3 pli respectively. Due to the closeness of the results, and the lack of an apparent plateau, no WOT estimates could be accurately made for these graphs. As seen in the graphs, changing the differential torque seems to have little effect of the radial pressure. When comparing these graphs to the repeatability graph, all pressures fell roughly within the 5 psi error range. Therefore, for these winding conditions, the difference in torques seems to have little effect on WOT. Note that by only changing torque differential the total torque is held constant.

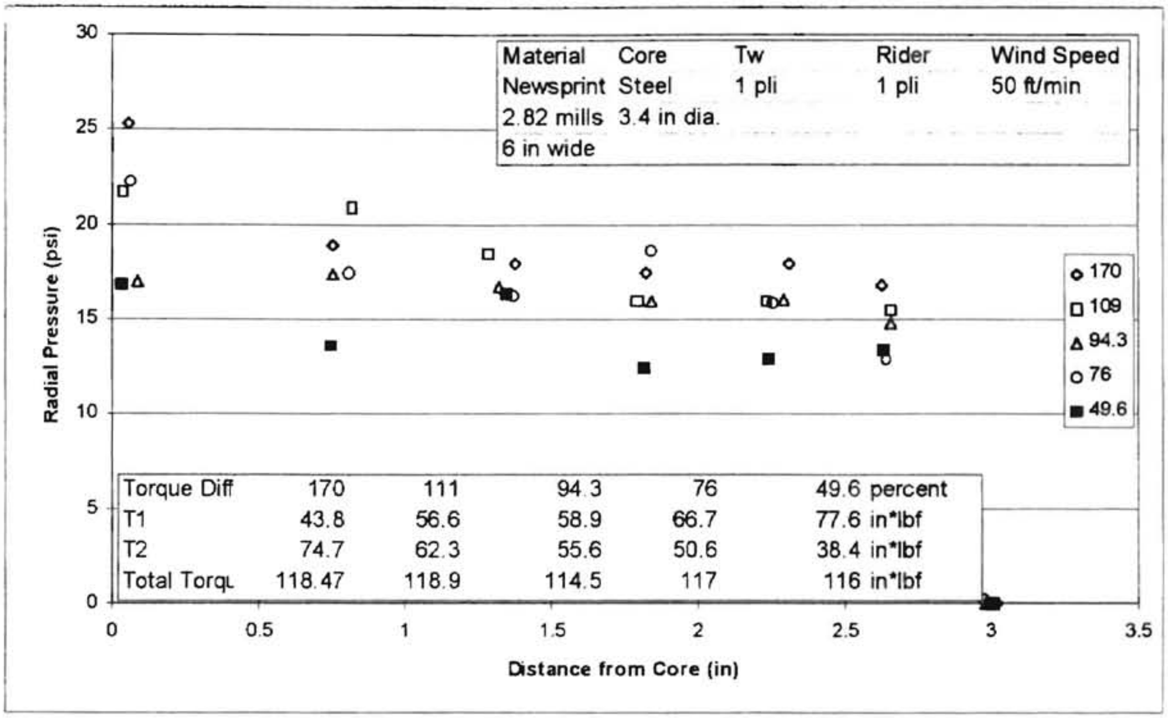

Figure 16: Effect of Differential Torque for Rider Load at 1 pli, Tw 1 pli.

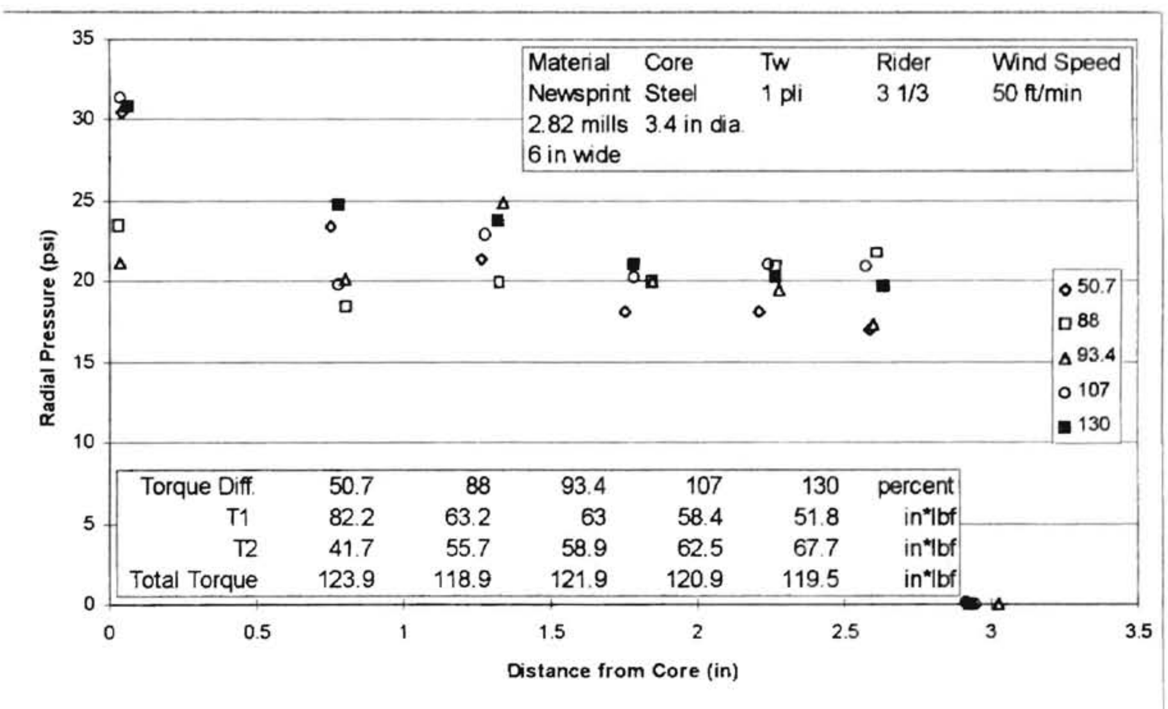

Figure 17: Effect of Differential Torque for Rider Load at 3 1/3 pli, Tw at 1 pli.

Figure 18 shows the results of an experiment of the effect of torque differential with higher drum torques. For this experiment the incoming web tension was increased to 3 pli, and the rider roll load remains at 3 1/3 pli. By increasing the web line tension, both torques on each drum dramatically increase. Here it is seen that increasing torque difference, or redistributing more torque to drum two, does indeed give increasingly higher radial pressures. Background research, such as that given by Good, Wu, and Fikes[3] equation [15], indicates increasing Tw increases the effect on WOT. For two-drum winding, increasing web line tension requires higher torques to be produced by the two drums. With higher torques produced, changing the torque difference also increases the radial pressures in the rolls. Thus of the four winding variables in two-drum winding , Tw, T_1, and T_2 cannot be independently changed.

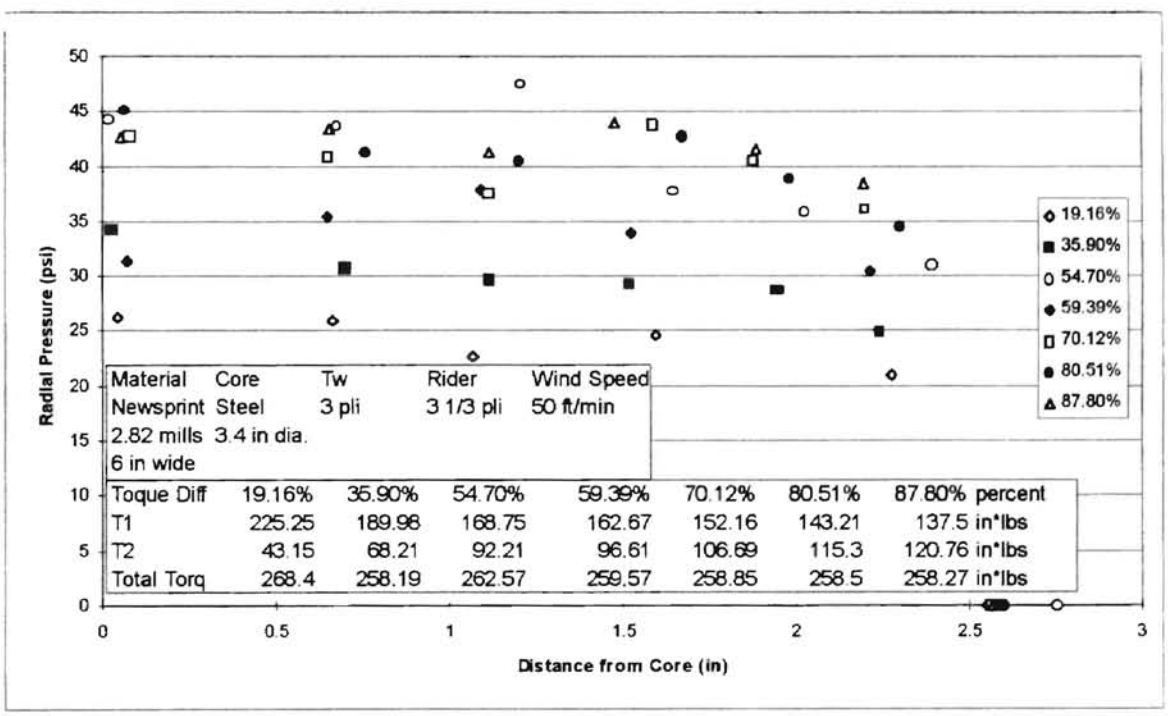

Figure 18: Effect of Torque Differential with Rider Load at 3 1/3 pli, Tw at 3 pli

Figure 19 focuses on how changing the web line tension affects the radial pressure. By increasing web line tension it can be seen that the total torque also increases from roll to roll. As seen in the graph, increasing Tw 1 pli causes roughly an increase of 10 psi in radial pressure. The torque differential for each one of the rolls was originally set at 100 percent. However, as can be seen the torque difference was not met, and the given torque difference is based on the measured torques. As also can be seen, as Tw increases the torque difference decreases. Previous experiments show that as torque difference decreases radial pressure would also decease, and this must be taken into account in WOT predictions.

As discussed before, increasing Tw above 4.4 causes the electric motors to reset due to a torque overload. It is believed that the radial pressure profile for Tw of 4.4 pli is a bad test roll due to a reset of the motor. It was immediately noticed that the motor had shut off, and therefore, the motor was immediately reset. Due to the difficulty in getting a higher then 4 pli test run, and the shortness of the motor shutdown this 4.4 pli test run was kept. As can be seen from Figure 19, this 4.4 pli test run has a much lower radial pressure profile then the 4 pli case. Therefore, for the above mentioned reason, the 4.4 pli roll is believed to be a bad test run.

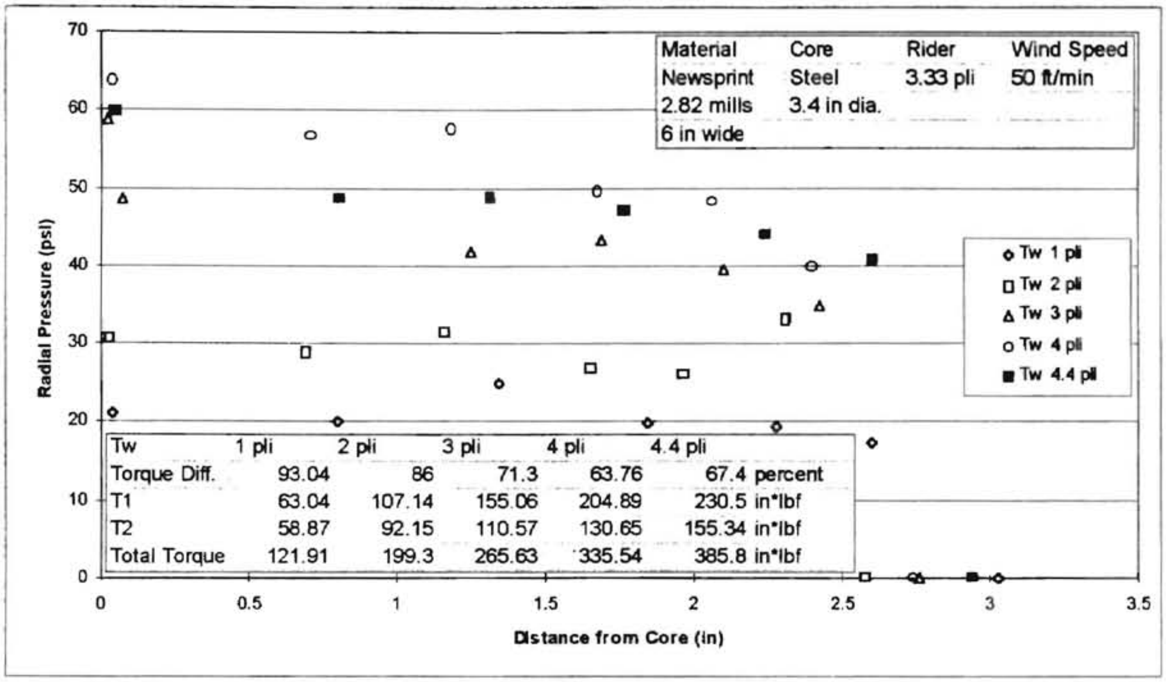

Figure 19: Effect of Changing Web Line Tension

Figure 20 and Figure 21 show the effects of changing torque differential with higher rider roll loads of 6 2/3 and 10 pli. It was desired to learn how the rider roll load would affect the radial pressure profile. For a rider roll load of 6 2/3 pli, the incoming web tension was at 3 pli(See Figure 20). In this graph, it can be seen that increasing the torque differential increases the radial pressure. For a rider roll load of 10 pli the incoming web tension was set at 1 pli(See Figure 21). For this graph, it can be seen that changing torque differential has little effect on radial pressure as was the case previously. Also, notice that the overall radial pressures are much higher than those at rider roll loads of 3 1/3 pli.

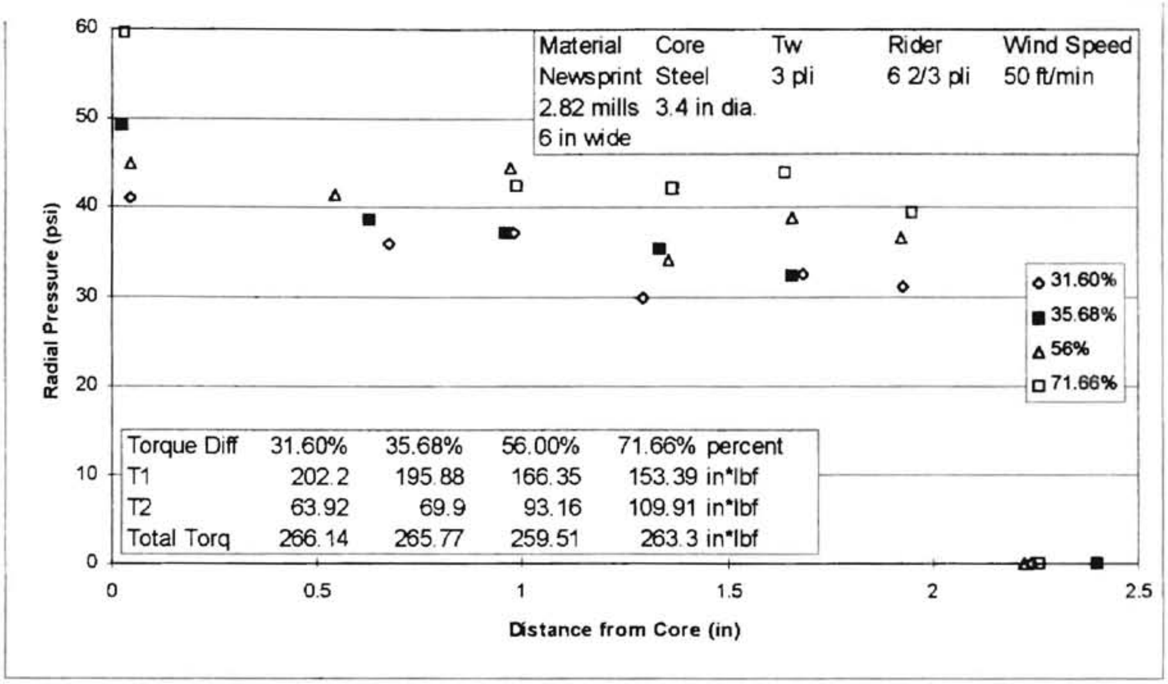

Figure 20: Effect of Differential Torque Rider of 6 2/3 pli

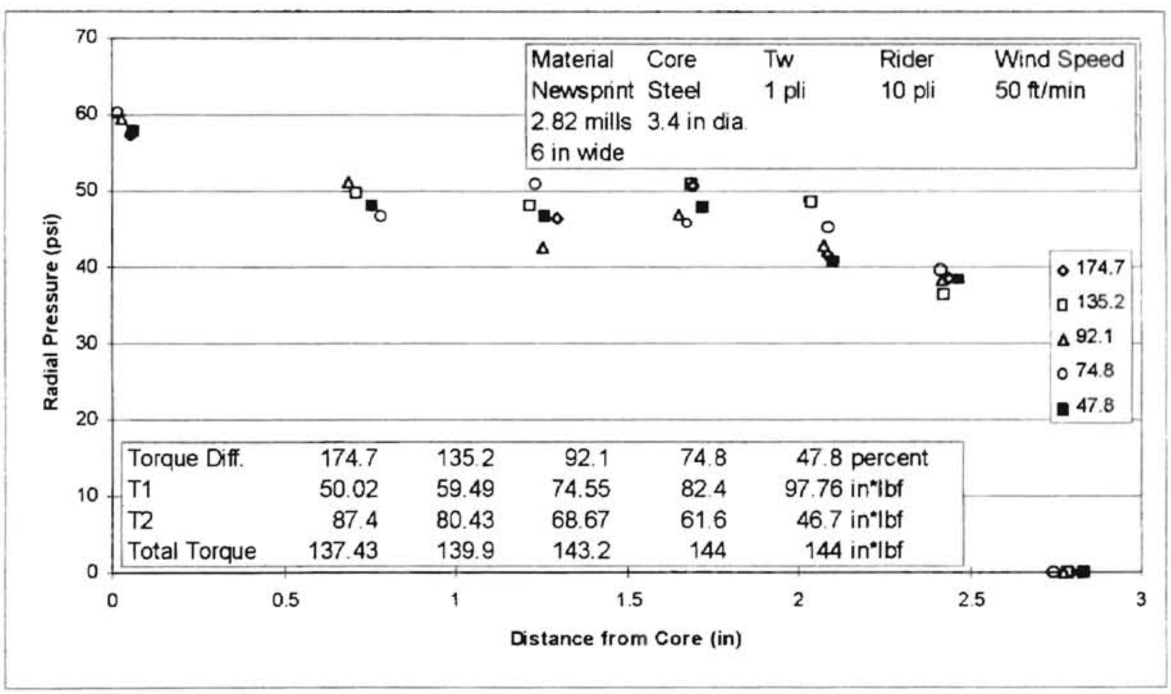

Figure 21: Effect of differential Torque for Rider of 10 pli

Two-Drum WOT Predictions

From Figure 18 through Figure 21 WOT predictions could be made as discussed previously. These estimated WOT predictions are presented in Table 2.

Figure #	T1(in*lbs)	T2(in*lbs)	Tw(pli)	Rider Load	Average Press. Profile	Est. WOT
18	225.25	43.15	3	3.33	24.9	746
18	189.98	68.21	3	3.33	30.5	828
18	162.67	96.61	3	3.33	34.7	886
18	152.16	106.69	3	3.33	41.1	967
18	137.5	120.76	3	3.33	42.6	985
19	63.04	58.87	1	3.33	21.1	684
19	107.14	92.15	2	3.33	28.3	797
19	155	110	3	3.33	41.6	973
19	230.5	155.3	4.4	3.33	47.2	1039
19	204.9	130.6	4	3.33	53.1	1105
20	202.2	63.92	3	6.66	33.3	867
20	195.88	69.9	3	6.66	35.9	901
20	166.35	93.16	3	6.66	40	954
20	153.4	109.9	3	6.66	42	978
21	74.5	68.7	1	10	44.2	1004
21	50	87.4	1	10	46.2	1028
21	82.4	61.6	1	10	47.2	1039
21	59.5	80.4	1	10	49.2	1062

Table 2: Two-Drum WOT Predictions

Torque Minus Web line Tension

It is apparent that web line tension and drum torques are related, and an interesting note can be made in regard to the torque minus web line tension. If the torque produced by incoming web line tension, Tw, on drum one is subtracted from the total torque, a constant number remains for each roll wound in a given experiment (See Table 3). This can be done by multiplying Tw by drum radius of 12 inches, and the width of the winding roll of 6 inches. This torque required to overcome web tension is then subtracted from the total torque where there remains some 40 to 70 inch pounds of torque. The remaining torque would of course change if the rider roll load were changed due to rolling resistance. Therefore, rolling resistance is then subtracted from the remaining amount. Afterward, some 20 through 37 inch pounds of torque remains that must somehow contribute to the WOT. Given that torque measurements vary by ± 5 inch pounds, it is noted that the remaining torque is fairly constant from roll to roll. The roll that has a remaining 51.2 inch pounds of torque is the roll wound with Tw at 4.4 pli, this is the bad roll discussed previously.

Figure #	T1(in*lbs)	T2(in*lbs)	Tw(pli)	Rider Load	Rolling Resistance	Total Tor - Tw*72	Minus Roll Res.
16	43.8	74.7	1.0	1.0	15.0	46.5	31.5
16	56.6	62.3	1.0	1.0	15.0	46.9	31.9
16	58.9	55.6	1.0	1.0	15.0	42.5	27.5
16	66.7	50.6	1.0	1.0	15.0	45.3	30.3
16	77.6	38.4	1.0	1.0	15.0	44.0	29.0
17	51.8	67.7	1.0	3.3	17.8	47.5	29.7
17	58.4	62.5	1.0	3.3	17.8	48.9	31.1
17	63.0	58.9	1.0	3.3	17.8	49.9	32.1
17	63.2	55.7	1.0	3.3	17.8	46.9	29.1
17	82.2	41.7	1.0	3.3	17.8	51.9	34.1
18	137.5	120.8	3.0	3.3	17.8	42.3	24.5
18	152.2	106.7	3.0	3.3	17.8	42.9	25.1
18	162.7	96.6	3.0	3.3	17.8	43.3	25.5
18	190.0	68.2	3.0	3.3	17.8	42.2	24.4
18	225.3	43.2	3.0	3.3	17.8	52.4	34.6
19	63.0	58.9	1.0	3.3	17.8	49.9	32.1
19	107.1	92.2	2.0	3.3	17.8	55.3	37.5
19	155.0	110.0	3.0	3.3	17.8	49.0	31.2
19	204.9	130.6	4.0	3.3	17.8	47.5	29.7
19	230.5	155.3	4.4	3.3	17.8	69.0	51.2
20	153.4	109.9	3.0	6.7	26.7	47.3	20.6
20	166.4	93.2	3.0	6.7	26.7	43.5	16.8
20	195.9	69.9	3.0	6.7	26.7	49.8	23.1
20	202.2	63.9	3.0	6.7	26.7	50.1	23.4
21	50.0	87.4	1.0	10.0	45.3	65.4	20.1
21	59.5	80.4	1.0	10.0	45.3	67.9	22.6
21	74.5	68.7	1.0	10.0	45.3	71.2	25.9
21	82.4	61.6	1.0	10.0	45.3	72.0	26.7

Table 3: Torque Minus Web Line Tension

SUMMARY OF RESULTS

One of the most prominent trends in the data is the effect of web line tension seen in Figure 19. Note that web line tension and total winding torque are inseparable and so one might be drawn to conclude that:

- Web line tension is an important variable in determining WOT

or

- Total winding torque is important in determining WOT.

There seems to be mixed results concerning nip load. For cases where Tw = 1 pli at roughly 93 percent difference in torque, we have:

Difference in Torque (%)	93.4	93.04	92.1
Rider Loads (pli)	1	3 1/3	10
Plateau Pressure (psi)	17	21.1	44.2

Table 4: Comparison of Plateau Pressures for Tw of 1 pli.

and so the plateau pressure and thereby the WOT seem to be effected by rider load level. However, when Tw = 3 pli at roughly 70 percent difference in torque we have:

Difference in Torque (%)	71.3	71.66
Rider Load (pli)	3 1/3	6 2/3
Plateau Pressure (psi)	41.6	42

Table 5: Comparison of Plateau Pressures for Tw of 3 pli.

and thus the plateau pressure and WOT seem insensitive to rider load level.

There are also mixed results for the difference in torques percent. Figure 16 shows

$\pm$ 2.5 psi variation in plateau pressure for a 50% to 170% difference in torque. Since this falls within the repeatability range displayed in Figure 15 it must be concluded that the percent difference in torque is a minor effect. The results shown in Figure 17 are similar. However, in Figure 18 and Figure 20 a 20 psi range and a 10 psi range in plateau pressure can be seen as a function of torque difference, definitely above the $\pm$ 2.5 psi repeatability range. Minor effects are shown in Figure 21. Now, Figure 18 and Figure 20 are compared where the web line tension is held at 3 pli (See Table 6).

Figure 18				Figure 20			
Rider load 3 1/3				Rider Load 6 2/3			
% Torque	35.9	59.39	70.12	% Torque	35.68	56	71.66
Plateau Press. (psi)	30.5	34.7	41.1	Plateau Press. (psi)	35.9	40	42
Total Torque (in lbs)	258	259.57	258.85	Total Torque (in lbs)	265.77	259.51	263.3

Table 6: Comparison of Plateau Pressure for Differing Rider Roll Loads.

The nip load is doubled while all else remained constant and the question to asked "Are the plateau pressures different?" Given an error range of $\pm$ 2.5 psi, it would seem for the two lower percent torque differences that doubling the nip load does increase the plateau pressures, but at the highest percent torque differences, the plateau pressures are not significantly different.

It was noted in the experiments that increasing the web line tension did increase the radial pressures. However, increasing the web line tension also has the effect of naturally increasing the torques supplied to the winding drums. A relation has been found in regard to the total torque minus the incoming web line tension. When calculations of total torque minus web line tension and rolling resistance were made, 20 to 37 inch pounds of torque remained for most of the rolls. Given that $\pm$ 5 inch pound torque measurement error occurs, this remaining torque appears to be constant from roll to roll.

CHAPTER 6

DISCUSSION

The wound-on-tension, WOT, was found for various winding conditions in chapter 5. In this chapter an empirical model will be developed relating WOT to the winder variables. This model is partially developed upon previous background research and from the four winding variables in two-drum winding. This model is then compared to the measured results given in chapter 5. Also, an equation will be derived from Rand and Eriksson's polar plots[7], and Olsen's two-drum equation[10] will be compared to the measured results.

SURFACE WINDING WOT

As discussed in earlier, two-drum winding is similar in nature to surface winding. The surface winding model is:

$$WOT = \mu_{k,p/p} * \frac{N}{h}$$

[19]

where N is the load to the Nip roller, h is the web thickness, $\mu_{k,p-p}$ is kinetic coefficient of friction of paper to paper. As seen in the results in chapter 5, increasing rider roll load increases the WOT. The same can be seen in the surface-winding model by increasing nip roller load, N. However, due to two-drum geometry, as the winding roll gets bigger, the weight of the roll becomes a factor, and the normal reaction force to the roll changes, Figure 22.

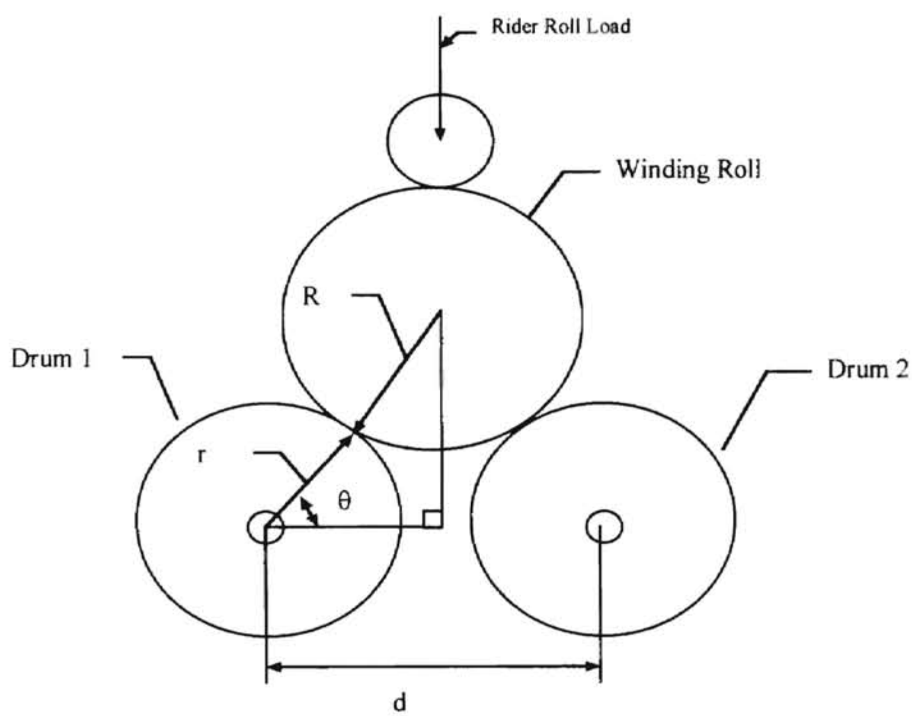

Figure 22: Geometry Change in Two Drum Winding

Of primary interest is the normal force due to drums one and two. The reaction force, N, of drum one changes as a function of radius of the winding roll given by:

$$\theta = \cos^{-1}(d / (2 * (r + R))) \tag{20}$$

$$N = \frac{(Rider\ Load + Core\ \text{Weight} + \text{Paper Weight})}{2\sin\theta}$$

[21]

Where R is the radius of the winding roll, r is the radius of the winding drums, and θ is the angle generated by geometry. The Rider Load is the load on the rider, and the Core Weight and Paper Weight are the respective weights of said items affecting the normal reaction force.

If it is assumed that both drums in the two-drum winding has a surface winding effect on equation **[21]**, and that nip WOT effects are additive, then the reaction force, N, becomes:

$$N = \frac{(Rider\ Load + Core\ \text{Weight} + \text{Paper Weight})}{\sin\theta}$$

[22]

This N, from equation [22], is then substituted into the surface winding model of [19].

With the surface WOT equation developed, Table 7 is produced to give some idea of range of the normal reaction force on drum one for this machines winding setup. This normal reaction force equation is comprised of the rider roll load and the weight of the paper. Upon startup of winding, the rider roll load will produce most of the normal reaction force. With roll build up, the weight of the paper will produce most of the normal reaction force. However, as the roll becomes bigger the angle theta increases, thereby decreasing the normal reaction force. If the startup radius of the roll, the outside radius of the core, is 1.7 inches and the finial radius of the roll is 4.7 inches, then theta becomes 26.6 and 42.8 degrees respectively.

Rider Load (pli)	Core Weight lbs	Paper Weight (lbs) Angle Theta (deg) 26.6 (deg)	42.8 (deg)	N (pli) Angle Theta (deg) 26.6 (deg)	42.8 (deg)
1	3.5	0	6.5	1.77	1.96
3 1/3	3.5	0	6.5	4.37	3.68
6 2/3	3.5	0	6.5	8.10	6.13
10	3.5	0	6.5	11.82	8.59

Table 7: Two-Drum Normal Reaction Force Estimates

An interesting case develops here, if the nip WOT effects are based upon the nip with the highest load, then in some cases the rider load will determine the WOT of the roll. As can be seen from Table 7, for most cases the normal reaction force will produce most of the WOT, except for the cases of 6 2/3 and 10 pli at 42.8 degrees. For these cases the rider roll load having the highest load will produce the WOT. This above mentioned effect is neglected in the rest of the calculations in this report. This effect will only take affect towards the very end of the winding process, outside the plateau pressure range. Because, only the plateau pressure range is used for calculations in this report, then the above effect is not seen in any of the calculations.

In the model just developed, equation [22], has some interesting properties that have not shown up in the test results. Because of machine limitations, no rolls with more then 3 inches of pile height of paper could be wound. In industry when a roll reaches a certain size, the WOT becomes so great from the weight of the roll that the web line breaks. For experiments in this project, this limit was never reached. To give some idea of what a pressure profile may look like for rolls wound greater then 3 inches of pile height, Figure 23 was produced based on equation [19] and [22] plus a constant.

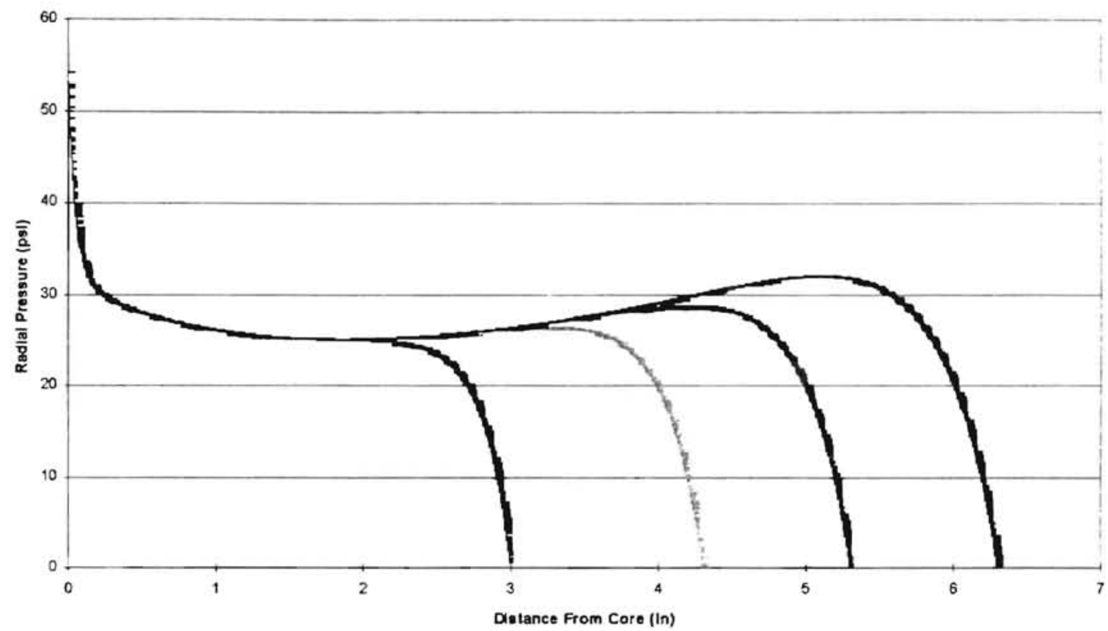

Figure 23: Predictions of Surface Model for Larger Radius Rolls

As seen in the graphs for rolls greater then 4 inches of pile height, the radial pressure increases as a function of the distance from the core. This is because, as the roll gets bigger, more weight is added to the roll thus increasing the WOT. Also, rolls that have 3 inches of pile height have plateaus similar to those seen in the experimental results.

DERIVATION OF AN EXPERESSION FOR WOT

With WOT the predictions made for the winding cases, presented in Table 2, a two-drum winding empirical model is developed. It was assumed for this project that the model will be of a form of the four operating parameters: web tension, rider load, torque 1, torque 2 and the WOT due to surface winding effects by [19] and [22]. The WOT formula would be of the form:

$$
\begin{aligned}
&WOT = \\
&C_1 Surface \quad WOT + C_2 T_1 + C_3 T_2 + C_4 Tw + \\
&C_5 Rider \quad Load + C_6 T_1^2 + C_7 T_2^2 + C_8 T_1 T_2 + C_9
\end{aligned}
\qquad [\,23]
$$

Where equation [23] assumes that multiplies of the torques may occur and a constant may remain.

To solve for the constants in equation [23], all relevant data from the four graphs with WOT predictions were reproduced in Table 8.

Figure #	Finial Radius	Estimated WOT(psi)	Surface WOT(psi)	Estimated h (mills)	Estimated WOT(pli)	Surface WOT(pli)	T1(in*lbs)	T2(in*lbs)	Tw(pli)	Rider Load	Predicted WOT	ABS Diff.	Below 141(pli)	Percent Error
19	4.73	684	518	2.82	1.93	1.46	63.04	58.87	1	3.33	1.95	0.02	Yes	0.96
18	4.3	746	518	2.82	2.10	1.46	225.25	43.15	3	3.33	2.10	0.00	Yes	0.01
19	4.28	797	518	2.82	2.25	1.46	107.14	92.15	2	3.33	2.52	0.27	No	12.01
18	4.3	828	518	2.82	2.34	1.46	189.98	68.21	3	3.33	2.19	0.14	Yes	6.11
20	3.94	867	916.5	2.82	2.44	2.58	202.2	63.92	3	6.66	2.50	0.05	Yes	2.09
18	4.3	886	518	2.82	2.50	1.46	162.67	96.61	3	3.33	2.50	0.01	Yes	0.24
20	4.1	901	916.5	2.82	2.54	2.58	195.88	69.9	3	6.66	2.55	0.01	Yes	0.38
20	3.94	954	916.5	2.82	2.69	2.58	166.35	93.16	3	6.66	2.69	0.00	Yes	0.00
18	4.3	967	518	2.82	2.73	1.46	152.16	106.69	3	3.33	2.60	0.13	Yes	4.64
19	4.46	973	518	2.82	2.74	1.46	155	110	3	3.33	2.74	0.01	Yes	0.32
20	3.94	978	916.5	2.82	2.76	2.58	153.4	109.9	3	6.66	2.92	0.17	No	6.01
18	4.3	985	518	2.82	2.78	1.46	137.5	120.76	3	3.33	2.74	0.04	Yes	1.53
21	4.5	1004	1316	2.82	2.83	3.71	74.5	68.7	1	10	2.83	0.00	Yes	0.00
21	4.5	1028	1316	2.82	2.90	3.71	50	87.4	1	10	2.93	0.03	Yes	1.10
21	4.5	1039	1316	2.82	2.93	3.71	82.4	61.6	1	10	2.77	0.16	No	5.40
19	4.64	1039	518	2.82	2.93	1.46	230.5	155.3	4.4	3.33	3.72	0.79	No	27.05
21	4.5	1062	1316	2.82	2.99	3.71	59.5	80.4	1	10	2.90	0.10	Yes	3.19
19	4.44	1105	518	2.82	3.12	1.46	204.9	130.6	4	3.33	3.06	0.05	Yes	1.73

Constants	C1	C2	C3	C4	C5	Total
	0.500	0.016	0.027	-1.041	-0.104	1.974
	C6	C7	C8	C9		
	0	0	0	0		

Table 8: Empirical Analysis of WOT Estimates

In the table, on the second column is the final roll radius of a given roll, and the third column is the estimated WOT derived earlier. The column labeled surface winding is the estimated average WOT that may occur from surface winding effects of equation [19] and [22]. For this table, it was assumed that web thickness, h, was 2.82 mils. This h value is then used to convert values of WOT in psi to pli. Columns labeled T1, T2, Tw, and Rider Load are the four winding variables measured during winding that where shown in the previous graphs.

The predicted WOT is equation [**23**] produced from the constants given at the bottom of the table, and the various winding variables. The column labeled "Below 0.141(pli)" is meant to give some measure of accuracy to the predicted WOT. A close prediction in WOT should be within 50 psi of the estimated WOT, or assuming h is 2.82 mils, 0.141 pli. If these criteria are met then this column is marked "Yes." The percent error column is the percent difference of the predicted WOT to that of the estimated WOT. A good approximation of the WOT was met if the error is below 5 percent. An almost perfect approximation was met if the error is less than 2 percent. Percent errors much above 10 percent were reasonable close given the potential for errors between pull-tab measurements, and varying material properties of paper , but are considered inaccurate.

With all the data entered into a spreadsheet, the constants were solved for using the built in solver in the spreadsheet. The solver program was set up so that one cell could be brought to a minimum value while changing the cells that contain the constants.

The absolute difference column, the difference between the predicted WOT and estimated WOT, was summed up and that value is given by the total. The value by total was the desired cell to be made a minimum value by the solver. The above constants were answers provided by the solver after the solver was set up.

The solution to the constants above could be varied somewhat and still provide percent errors below 5 percent. For example, if C1 can be set to the assumed value of 1, the rest of the constants can be resolved. If the percent errors remain below 5 percent, this value for C1 is workable. Then, another value for another constant can be picked and the remaining constants can be resolved. If the percent error is acceptable again, these values are workable. This procedure was used to derive a theoretical model for two-drum winding. Only instead of using random numbers, estimations from background research, or machine dimensions, such as radius of drums, were used.

Some of the estimations were as follows. For the surface winding WOT, it was assumed that both drums would contribute to the WOT. Therefore, possible constants for surface WOT would be either 0, 0.5, or 1. Zero would be given if there were no effect, while 0.5 would be the case if only one drum had an effect. Likewise, if both drums affected WOT, the value would be one. The torques constant would most likely to be a function of the radius of the drums and the width of the roll. Therefore, the most likely constant for torques would be 1/72 of either sign. For the incoming web tension, the assumed values could be zero, or one if the full effect of Tw is seen or not. If however the exponential effect of equation [4] were seen, this value would be closer to 0.58. The wrapped capstan is estimated at roughly 0.69 π radians with a static friction value of 0.25. The effect of the rider roll could be zero, or from the effect of equation [9] the constant value would be the kinetic friction constant of 0.19. For constants C6 through C9, it is unknown how these constants will effect WOT if any.

After adjusting Table 8 according to the estimations the following assumptions about the WOT predictions can be made. For all tries, constants C6 through C9 can always be set to zero with little changes in percent error. Therefore, no affects of the torques multiplied to themselves are seen. None of the other constants could be made to come to zero without creating percent errors above 15 percent for a portion of the predictions. The exact value of the remaining constants were subject to some interpretation. Only one, or possible two of the remaining estimations could be met without creating large percent errors. As a note, for almost all cases tried, T_2 usually had constant values, C3, which were twice that of C2. Setting C2 and C3 to the same value of either sign also produced unacceptably high percent errors.

The above procedure assumes that the percent error of WOT should be minimized. However, given that material properties vary from roll to roll combined with other potential measurement errors, acceptable values for percent error could be as high as 10 to 15 percent. In other words, more of the estimations could be met if higher percent errors were allowed. By not being able to change things like the radius of the winding drums, and winding larger rolls, the true relationships of the constants could not be determined. At this point, relating the constants to drum radius etc., could not be proved and would provide unwarranted higher errors in WOT predictions. Therefore, to give some idea for magnitude and to keep some accuracy, C1 was set to 0.5, and C6 through C9 were set to zero. The remaining values are those found by the solver. These were the constants presented in Table 8, and equation [24] was derived from this table.

$$WOT = \frac{1}{2} Surface\ WOT + \frac{1}{62.5} T_1 + \frac{1}{37.0} T_2 - 1.041 Tw - 0.104 Rider\ Load$$

[24]

This expression [24] is an empirical formula. As such, no real physical significance can be derived from this formula. The constant values for the incoming web tension and that of the rider roll are negative in value, which is contrary to previous experimental results. For example, the rider roll load is seen to increase WOT in some experiments, therefore, it would be natural to assume this constant value should be of positive value. Likewise, the same argument could be made for that of the sign of the incoming web tension. What this empirical formula should prove is that the above variables are the most likely contributors to the WOT for two-drum winding.

COMPARISON WITH RAND AND ERIKSSON

As discussed in the chapter Literature Review, Rand and Eriksson produced a polar plot of the WOT for the two-drum winder (See Figure 3 and Figure 4). From these plots, assumptions can be made regarding nip mechanics and compared to the WOT estimates. As seen from the plots the rider roll and drum two contribute most of the WOT for the roll. If this is the case, then nip-induced effects due to drum one must be negligible. Also, seen in the graphs as the winding roll becomes larger the angle of the normal reaction forces to the rolls changes, and the WOT becomes larger. This effect is taken into consideration by the surface winding effects of the two-drum winder from equations [19] and [22]. With the above facts in mind, Table 9 was reproduced from Table 2.

The surface winding constant, C_1, was set for ½ to meet the above requirements. The constant for T_2, C_3, was set to 1/72, which is the radius of the drums times the width of the roll. The solver constantly gave the rider roll constant a negative value, which goes

Figure #	Finial Radius	Estimated WOT(psi)	Surface WOT(psi)	Estimated h (mills)	Estimated WOT(pli)	Surface WOT(pli)	T1(in*lbs)	T2(in*lbs)	Tw(pli)	Rider Load	Predicted WOT	ABS Diff.	Below .141(pli)	Percent Error
19	4.73	684	518	2.82	1.93	1.46	63.04	58.87	1	3.33	1.67	0.26	No	13.35
18	4.3	746	518	2.82	2.10	1.46	225.25	43.15	3	3.33	1.70	0.40	No	19.12
19	4.28	797	518	2.82	2.25	1.46	107.14	92.15	2	3.33	2.26	0.01	Yes	0.47
18	4.3	828	518	2.82	2.34	1.46	189.98	68.21	3	3.33	2.05	0.29	No	12.30
20	3.94	867	916.5	2.82	2.44	2.58	202.2	63.92	3	6.66	2.55	0.11	Yes	4.33
18	4.3	886	518	2.82	2.50	1.46	162.67	96.61	3	3.33	2.44	0.05	Yes	2.19
20	4.1	901	916.5	2.82	2.54	2.58	195.88	69.9	3	6.66	2.63	0.09	Yes	3.61
20	3.94	954	916.5	2.82	2.69	2.58	166.35	93.16	3	6.66	2.96	0.27	No	9.96
18	4.3	967	518	2.82	2.73	1.46	152.16	106.69	3	3.33	2.58	0.14	Yes	5.29
19	4.46	973	518	2.82	2.74	1.46	155	110	3	3.33	2.63	0.12	Yes	4.20
20	3.94	978	916.5	2.82	2.76	2.58	153.4	109.9	3	6.66	3.19	0.43	No	15.64
18	4.3	985	518	2.82	2.78	1.46	137.5	120.76	3	3.33	2.78	0.00	Yes	0.00
21	4.5	1004	1316	2.82	2.83	3.71	74.5	68.7	1	10	2.93	0.10	Yes	3.57
21	4.5	1028	1316	2.82	2.90	3.71	50	87.4	1	10	3.19	0.30	No	10.18
21	4.5	1039	1316	2.82	2.93	3.71	82.4	61.6	1	10	2.83	0.10	Yes	3.26
19	4.64	1039	518	2.82	2.93	1.46	230.5	155.3	4.4	3.33	3.43	0.50	No	17.09
21	4.5	1062	1316	2.82	2.99	3.71	59.5	80.4	1	10	3.10	0.10	Yes	3.40
19	4.44	1105	518	2.82	3.12	1.46	204.9	130.6	4	3.33	3.04	0.08	Yes	2.45

Constants	C1	C2	C3	C4	C5	Total	3.34
	0.500	0.000	0.0139	0.124	0.000		
	C6	C7	C8	C9			
	0	0	0	0			

Table 9: Comparison of Rand and Eriksson Plots to WOT Estimates

against positive WOT increase seen in the polar plots. Also, if it is assumed that if only the nip with the highest load producing the most WOT should only be considered, then the above assumption would prove to be correct. For example, the normal force to drum two is in most cases higher then the normal force of the rider roll.

Therefore, if it is assumed that the highest normal force determines the WOT, then drum two would produce the WOT. Therefore, the rider roll constant should be zeroed out. The remaining constant Tw is determined by the solver. Table 9 is the derived results from these assumptions. The percent error results for these assumptions are found to be within acceptable range. The derived formula form Table 9 then is given by the equation:

$$WOT = 0.5 * Surface\ WOT + \frac{1}{72} T_2 + 0.124 * Tw \qquad [25]$$

COMPARISON WITH OLSEN'S FORMULA

Olsen[9] derived an analytic model [11] for the two-drum winder that included inertial effects which is now compared to the measured radial pressures profiles given in the Results section of this report. If it is assumed that velocity of the winding roll and rolling resistance is negligible, Olsen's[9] formula becomes:

$$WOT = \frac{1}{2}\left(Tw + \frac{M_2}{r_2} - \frac{M_1}{r_1}\right) + F_n \qquad [26]$$

where Fn is the surface winding effects given by equation [19] and [22]. From this

Figure #	Finial Radius	Estimated WOT(psi)	Surface WOT(psi)	Estimated h (mills)	Estimated WOT(pli)	Surface WOT(pli)	T1(in*lbs)	T2(in*lbs)	Tw(pli)	Rider Load	Predictec WOT	ABS Diff.	Below 141(pli)	Percent Error
19	4.73	684	518	2.82	1.93	1.46	63.04	58.87	1	3.33	1.20	0.73	No	37.72
18	4.3	746	518	2.82	2.10	1.46	225.25	43.15	3	3.33	0.97	1.14	No	54.06
19	4.28	797	518	2.82	2.25	1.46	107.14	92.15	2	3.33	1.63	0.62	No	27.62
18	4.3	828	518	2.82	2.34	1.46	189.98	68.21	3	3.33	1.38	0.95	No	40.72
20	3.94	867	916.5	2.82	2.44	2.58	202.2	63.92	3	6.66	1.83	0.61	No	25.07
18	4.3	886	518	2.82	2.50	1.46	162.67	96.61	3	3.33	1.77	0.73	No	29.07
20	4.1	901	916.5	2.82	2.54	2.58	195.88	69.9	3	6.66	1.92	0.62	No	24.58
20	3.94	954	916.5	2.82	2.69	2.58	166.35	93.16	3	6.66	2.28	0.41	No	15.06
18	4.3	967	518	2.82	2.73	1.46	152.16	106.69	3	3.33	1.91	0.81	No	29.79
19	4.46	973	518	2.82	2.74	1.46	155	110	3	3.33	1.92	0.83	No	30.11
20	3.94	978	916.5	2.82	2.76	2.58	153.4	109.9	3	6.66	2.49	0.27	No	9.71
18	4.3	985	518	2.82	2.78	1.46	137.5	120.76	3	3.33	2.11	0.66	No	23.91
21	4.5	1004	1316	2.82	2.83	3.71	74.5	68.7	1	10	2.32	0.52	No	18.25
21	4.5	1028	1316	2.82	2.90	3.71	50	87.4	1	10	2 62	0.28	No	9.75
21	4.5	1039	1316	2.82	2.93	3.71	82.4	61.6	1	10	2.21	0.72	No	24.54
19	4.64	1039	518	2.82	2.93	1.46	230.5	155.3	4.4	3.33	2.41	0.52	No	17.82
21	4.5	1062	1316	2.82	2.99	3.71	59.5	80.4	1	10	2.50	0.49	No	16.48
19	4.44	1105	518	2.82	3.12	1.46	204.9	130.6	4	3.33	2.21	0.90	No	28.91

Constants	C1	C2	C3	C4	C5	Total	11.81
	0.500	-0.0069	0.0069	0.500	0.000		
	C6	C7	C8	C9			
	0	0	0	0			

Table 10: Comparison of Olsen to WOT Estimates

formula, Table 10 was reproduced from Table 2. For each variable in Olsen's[9] formula, the appropriate constant value was given in the table. The percent errors given for this formula were found to be high and unacceptable. Therefore, Olsen's[9] formula for a two-drum winder, assuming negligible velocity, does not match the estimated WOT found in this project.

CHAPTER 7

CONCLUSIONS

SUMMARY

The objective of this research was to determine how each of the four winding variables, incoming web line tension, the two drum torques, and the rider roll load affect the wound-on-tension for two-drum wound rolls. To achieve this objective, the two-drum winding machine was successfully set up for experimentation. A range of experiments were conducted, demonstrating what possible factors may affect the radial pressure profiles in wound rolls. These radial pressure profiles were then related to an estimated WOT through a graph produced by Hakiel's center winding model [1]. The estimated WOT were then used to derive an empirical WOT model based upon the four winding variables, incoming web line tension, rider roll load, and the two drum torques. This empirical formula represents which of these four winding variables contribute to WOT. This empirical equation is given by:

$$WOT = \frac{1}{2} Surface\ WOT + \frac{1}{62.5} T_1 + \frac{1}{37.0} T_2 - 1.041 Tw - 0.104 Rider\ Load \qquad [27]$$

Furthermore, assumptions were made based upon data seen Rand and Eriksson's[7] polar graph, and a second empirical formula was derived. This equation is given by:

$$WOT = 0.5 Surface\ WOT + \frac{1}{72} T_2 + 0.124 Tw$$

[28]

Both empirical formulas represents the degree in which each of the four winding variables contribute to WOT. Also, Olsen's formula[10], with the assumption of negligible velocity, was then compared with the estimated WOT, and found to be inaccurate.

ACHIEVEMENTS AND CONCLUSIONS

The two-drum winder was assembled, and instrumented for the four winding parameters including: incoming web line tension, rider roll load, and two drum torques.. Conclusions drawn from the analysis of the experiments conducted are as follows:

- Increasing web line tension increases WOT.
- Increasing web line tension also increases drum torques as well.
- At higher drum torques, increasing torque differential increases WOT.
- Increasing Rider roll load, in most cases, increases WOT.

Furthermore, an empirical model developed here is based upon these four winding parameters can accurately predict WOT within a 6 percent error.

FUTURE WORK

The models developed in this report are empirical models, and can only predict WOT for rolls wound on the two-drum winding machine used in this project. Future WOT formulas could be further developed by making geometry changes to this two-drum winding machine, and relating these geometry changes to changes in radial pressures. For example, a major problem with roll telescoping prevents rolls being wound larger than 10 inches in diameter. Redesigning the machine such that the winding roll can not move horizontally will prevent this telescoping. Once rolls bigger than 10 inches in diameter can be wound, the effects of roll weight can be seen in radial pressure profiles. Another change could be made to verify the relationship of the torques to WOT. This change would require totally redesigning the machine such that differing drum radii could be used. Then relations between these changing drum radii to changes in radial pressure profiles could be determined.

Other modifications to this two-drum winding machine are also necessary to improve control of the rider roll load. Two liner bearings are used to guide the rider roll, and cause two problems. The first problem is that these guides have too much friction associated with them. The second problem is that because only two guides are used, the rider roll may come down in a slanted position. These two problems cause an improper applied load to the winding roll. The rider roll should be redesigned, to prevent the above mentioned problems.

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APPENDIX A

MAKING OF PULL TABS

Pull-tabs used consisted of a steel strip, the tab, with an outside covering of brass shim stock. The steel pull-tab is 8.5 inches long, 0.5 inches wide, and one thousandth of inch thick. These tabs have tape on the ends shaped to form a hook as a means for pulling the tap with a force gage. The brass shim stock is six inches long, the width of the winding roll, and one thousandth of an inch thick. The brass is then folded over twice to hold the tab, and is used to provide a known friction surface for the tab. The inside of the brass shim stock and the outsides of the steel pull-tab were cleaned with rubbing alcohol to remove any films.

Calibration of Pull Tabs

In order for the pull tabs to be useful they have to be calibrated to find the relationship of the pulling force required to dislodge the tab to the pressure applied to the outside of the tab. Six to seven pull-tabs are placed in a 6x6 inch stack of paper sheets of 1 inches high. Then a known downward pressure is supplied on the stack of paper containing the tabs. All pull-tabs are then pulled with a steadily increasing force with a force gage. At the point which pull-tabs begins to move, or slip, the pulling force is recorded (See Table 11). This pulling force was measured three times for each gage every time the downward pressure is changed. This pressure was changed for the range of radial pressure that the pull-tab will see inside of a wound roll. The range of pressures used was 10, 20, 30 40, and 50 psi.

Tab Series F

Calibration Psi'	1	2	3	4	5	6
10	15.2	14.62	14.72	14.64	13.16	12.08
10	15.12	15.04	14.26	14.68	13.38	12.64
10	15.52	15.36	19.98	14.9	13.12	12.1
20	27.02	29.6	28.4	28.36	24.94	24.06
20	27.58	28.02	27.6	28.56	24.68	24.14
20	27.32	29.56	28.14	27.88	25.92	23.98
25	34.66	35.6	33.46	32.74	30.32	28.46
25	34.66	36.14	33	34.66	31.58	29.14
25	33.78	35.22	32.76	33.98	31.52	29.48
30	38.1	41.78	37.66	38.02	34.68	34.78
30	40.76	42.62	40.24	39.52	37.78	35.1
30	40.88	43.24	40.5	40.64	37.4	36.92
Slope	0.798552	0.725054	0.846386	0.802544	0.843787	0.862709
Intercept	-2.08104	-0.91249	-3.48704	-2.06258	-1.14411	-0.52478

Table 11: An Example of Tab Series Calibration

After the entire pressure range has been covered, a linear curve fit that relates the pulling force to the pressure is determined. The slope and intercept for each tab was determined as those given in Table 11.

Using Pull Tabs

Pull tabs were inserted into the wound roll by applying adhesive tape to the tab envelope and slapping on to the unwind roll during unwinding. Each roll wound for this project has 1,500 feet paper wound into them, and the tabs are evenly spaced ever 250 feet web length. After the roll has been wound, each of pull-tabs was again pulled three times with a force gage. Each measured pulling force was then substituted into that tabs calibration equation to find the radial pressure. All three of the radial pressures were then averaged to give the presented radial pressure.

APPENDIX B

HAKIEL'S MODIFIED CENTER WINDING MODEL

```
/********      Hakiel's Modified Center Winding Model *************/
/********      for Two Drum winding/


/*Program for Web Handling
/*By Randy Turner

/*Finds Pressure in winding rolls
/*using Hakiel's model/

#include <stdio.h>
#include <stdlib.h>
#include <math.h>


void main(void)
{

      /*Winding Constants*/
      double CORE=1.7, CORE_WEIGHT=0.5833, WIDTH=6.0, ROLL_WEIGHT=0.0;
      double h=0.00282; /*inches*/
      double RIDER=3.33, T1=230.5, T2=155.3, Tw=4.4,;
      double finial_radius=4.64, FRICTION=0.19;

      /*Two Drum Winder Constants*/
      double Rad_drum=12;     /*Radius of Drums on winder (in)*/
      double Dis_drum=24.5;   /*Distance between drums on winder (in)*/


      /*Paper Properties*/
      double Et=490000;
      double Ec=80000;
      double Er[10]={0.0};
      double v=0.0;
```

```
/*Misc. Variables*/
double sinangle, *WOT, THETA;
double  *g_sq, *Press, p_one, p_two, radius, p_last, p_temp;
double A, B, C;
int WOL, total_lap, lap, i, j, num;


FILE *fp;
char file_name[20]="data.dat";


fp=fopen(file_name,"w");

/*Finding the total labs from radius of wound roll*/
total_lap=(finial_radius-CORE)/h;


/*Defining Er*/
Er[1]=40;


num=total_lap+1;

Press=calloc(num, sizeof(double));
if(Press==NULL)printf( "Press matrix did not initialize");

WOT=calloc(num, sizeof(double));
if(WOT==NULL)printf( "WOT matrix did not initialize");

g_sq=calloc(num, sizeof(double));
if(g_sq==NULL)printf( "g_sq matrix did not initialize");


/*Initializing To Zero*/
for(i=0;i<total_lap;++i)
{
      Press[i]=0.0;
      g_sq[i]=0.0;
}

/*For the first lap the P1 is(From B.C.)*/
radius=CORE;

/*Finding Wound on Tension*/
THETA=acos(Dis_drum/(2*(Rad_drum + radius)));
sinangle=sin(THETA);


/************************************************************/
/************************************************************/
/*******      Two Drum Winder Modification (WOT)     **********/
WOT[1]=
             ((RIDER + CORE_WEIGHT + ROLL_WEIGHT) * FRICTION)
             /(2*sinangle * h)
             +(T1/57.3
             +T2/35.6
             -1.13*Tw
             -0.109*RIDER)/h;

/************************************************************/
```

```
Press[1]=WOT[1]*h/radius;

/*For the Second lap(From B.C.)*/
radius=CORE+h;

/*Finding Wound on Tension*/
THETA=acos(Dis_drum/(2*(Rad_drum + radius)));
sinangle=sin(THETA);


/************************************************************/
/************************************************************/
/*******      Two Drum Winder Modification (WOT)    **********/
WOT[2]=
            ((RIDER + CORE_WEIGHT + ROLL_WEIGHT) * FRICTION)
            /(2*sinangle * h)
            +(T1/57.3
            +T2/35.6
            -1.13*Tw
            -0.109*RIDER)/h;

/************************************************************/

Press[2]=WOT[2]*h/radius;
Press[1]+=Press[2]/(h*((Et/Ec)-1+v)+1);

/*Strating at the second lap and the rest*/
/*WOL wound on Lap (N+1)*/
for(WOL=3;WOL<=total_lap;++WOL)
{

      /*Finding g*/
      for(j=1;j<WOL;++j)
      {
            g_sq[j]=Er[0];
            for(i=1;i<9;++i)
            {
                  /*Adding Er terms*/
                  g_sq[j]+=Er[i]*(pow(Press[j],i));
            }
            g_sq[j]=Et/g_sq[j];  /*Dividint Et by Er*/
      }


/************************************************************/

/************************************************************/
/*******      Two Drum Winder Modification (WOT)    *********/


      /*The Boundary Condition due to wound on tension*/
      radius=CORE+(WOL-1)*h;

      /*Finding Wound on Tension*/
      THETA=acos(Dis_drum/(2*(Rad_drum + radius)));
      sinangle=sin(THETA);
```

```
	/*Finding Current Weight of Roll*/
	ROLL_WEIGHT+=1.08/1500/12*2*3.14159*radius;

	WOT[WOL]=
		((RIDER + CORE_WEIGHT + ROLL_WEIGHT) * FRICTION)
		/(2*sinangle * h)
		+(-T1/144
		+T2/72
		+0.5*Tw)/h;

/************************************************************/

	p_last=WOT[WOL]*h/radius;


	/*Here I solve the matrix with respect to delta P1*/
	/*I increment p_one and p_two to represent diff press*/

	p_one=1;
	p_two=((Et/Ec)-1+v)*h+1;	/*In terms of p_one*/

	for(lap=2;lap<WOL;++lap)
	{
		radius=CORE+(lap-1)*h;

		/*Determing Matrix Constants*/
		A=1+3*h/(2*radius);
		B=(h*h)/(radius*radius)*(1-g_sq[lap])-2;
		C=1-3*h/(2*radius);

		/*Soliving for delta P1*/
		/*I put all of the matrix in terms of delta P1*/
		/*And solve it to the last B.C.*/
		/*Then Increment the p's to next pressures*/
		p_temp=p_two;
		if(lap!=(WOL-1))
		{
			p_two=-1*(p_two*B+p_one*C)/A;
			p_one=p_temp;
		}
		else
		{
			p_one=(-1*A*p_last)/(B*p_two+C*p_one);
		}

	}

	/*Knowing delta P[1] i now solve the matrix*/
	p_two=(((Et/Ec)-1+v)*h+1)*p_one;

	Press[1]+=p_one;
	Press[2]+=p_two;

	for(lap=2;lap<WOL;++lap)
	{
		radius=CORE+(lap-1)*h;
```

```
			/*Determing Matrix Constants*/
			A=1+3*h/(2*radius);
			B=(h*h)/(radius*radius)*(1-g_sq[lap])-2;
			C=1-3*h/(2*radius);
			p_temp=p_two;

			p_two=-1*(p_two*B+p_one*C)/A;
			p_one=p_temp;

			/*Adding delta P's to the total pressure*/
			Press[lap+1]+=p_two;
		}
	}

	/*Printing the Pressures*/
	for(lap=1;lap<=total_lap;lap=lap+10)
	{
		radius=CORE+(lap-1)*h;
		fprintf(fp, "\n %4d  %5.4f   %6.3f  %6.3f", lap,
			(radius-CORE), Press[lap], WOT[lap]);
	}
	fclose(fp);

}
```

APPENDIX C

LABVIEW PORGRAM SETUP

TWO_DRUM.VI
Last modified on 7/29/96 at 8:27 AM
Printed on 2/2/13 at 3:30 PM

Board Setup

Device

1

Channel drum 1

0

Channel drum 2

1

Torque Out Channel

0

Cal for Current

Current SF

171.00

Current Zero

50.00

Save File

c:\turner\data.dat

Torque Input Drum 2

50.00

Percent Torque Diff

1.00

Calibration

Drum 1 SF

555.00

Drum 1 Zero

120.00

Drum 2 SF

573.00

Drum 2 Zero

125.00

stop

STOP

Torque Graph

231.9
200.0
150.0
100.0
50.0
0.0
-50.0
-100.0
-150.0
-206.7

0 1:33

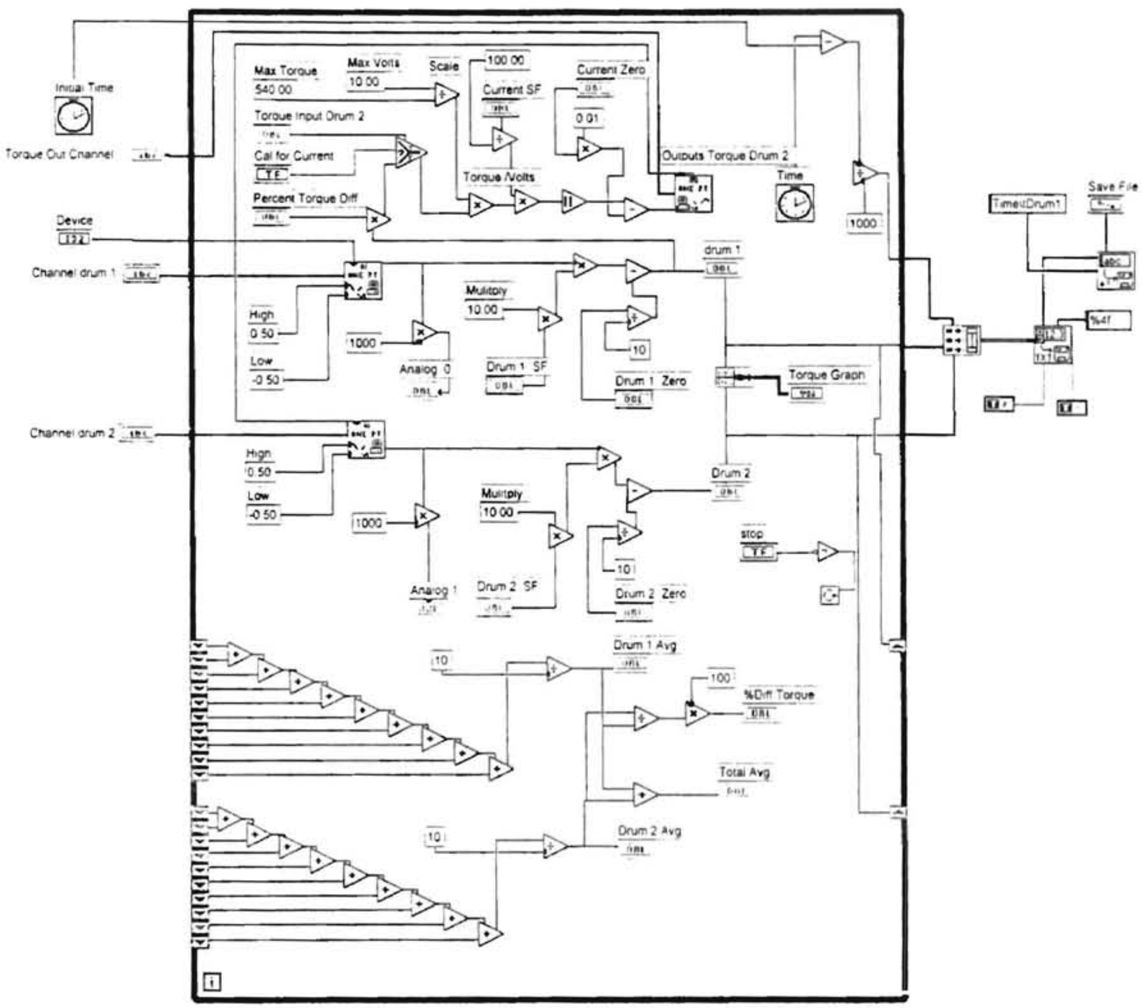
Drum 1
Drum 2

Data Input Area

Drum 1 Avg	drum 1	Analog 0
1.69	0.19	2.20
Drum 2 Avg.	Drum 2	Analog 1
-53.49	-50.27	-6.59
Total Avg	%Diff Torque	
-51.80	-3173.82	

TWO_DRUM.VI
Last modified on 7/29/96 at 8:27 AM
Printed on 1/31/13 at 10:44 AM

Block Diagram



VITA

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